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1893–1894,

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[Note.—The former Presidents of the Society are members of the Council for life or during their retention of active membership in the Society.]

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ART. 5. *Honorary Members*, not exceeding twenty-five in number, may be elected. They must be persons of acknowledged professional eminence who have virtually retired from practice.

ART. 6. To be eligible as an *Associate*, the candidate must have such a knowledge of, or connection with applied science as qualifies him, in the opinion of the Council, to co-operate with engineers in the advancement of professional knowledge.

ART. 7. To be eligible as a *Junior*, the candidate must have been in the practice of engineering for at least two years, or he must be a graduate of an engineering school.

The term "Junior" applies to the professional experience, and not to the age of the candidate. Juniors may become eligible to membership.

ART. 8. All Members and Associates shall be equally entitled to the privileges of membership. Honorary Members and Juniors shall not be entitled to vote nor to be members of the Council.

ELECTION OF MEMBERS.

ART. 9. Every candidate for admission to the Society, excepting candidates for honorary membership, must be proposed by at least three members, or members and associates, to whom he must be personally known, and he must be seconded by two others. The proposal must be accompanied by a statement in writing by the candidate of the grounds of his application for election, including an account of his professional experience, and an agreement that he will conform to the requirements of membership if elected.

ART. 10. All such applications and proposals must be received and acted upon by the Council at least thirty days before a regular meeting, when the Secretary shall at once mail to each member and associate, in the form of a letter ballot, the names of candidates recommended by the Council for election.

ART. 11. Any member or associate entitled to vote may erase the name of any candidate, and may, at his option, return to the Secretary such ballot enclosed in two envelopes, the inner one to be blank and the outer one indorsed by the voter.

ART. 12. The rejection of any candidate for admission as member, associate, or junior, by *seven* voters, shall defeat the election of said candidate. The rejection of any candidate for ad-

mission as honorary member by *three* voters shall defeat the election of said candidate.

ART. 13. The said blank envelopes shall be opened by the Council at any meeting thereof, and the names of the candidates elected shall be announced in the first ensuing meeting of the Society, and also in the first ensuing list of members. The names of candidates not elected shall neither be announced nor recorded in the proceedings.

ART. 14. Candidates for admission as honorary members shall not be required to present their claims; those making the nominations shall state the grounds therefor, and shall certify that the nominee will accept if elected. The method of election in other respects shall be the same as in case of other candidates.

ART. 15. All persons elected to the Society, excepting honorary members, must subscribe to the rules and pay to the Treasurer the initiation fee before they can receive certificates of membership. If this is not done within six months of notification of election, the election shall be void.

ART. 16. The proposers of any rejected candidate may, within three months after such rejection, lay before the Council written evidence that an error was then made, and if a reconsideration is granted, another ballot shall be ordered, at which thirteen negative votes shall be required to defeat the candidate.

ART. 17. Persons desiring to change the class of their membership shall be proposed in the same form as described for a new applicant.

FEES AND DUES.

ART. 18. The initiation fees of members and associates shall be $25, and their annual dues shall be $15, payable in advance. The initiation fee of juniors shall be $15, and their annual dues $10, payable in advance. A junior, being promoted to full membership, shall pay an additional initiation fee of $10. Any member or associate may become, by the payment of $200 at any one time, a life member or associate, and shall not be liable thereafter to annual dues. The Council shall have the power, by a unanimous vote at a regular meeting, to admit to the grade of life member, without the full payment of the sum above named, such person as for ten or more years has been a full member or associate, and who, in its judgment, is entitled to

this grade, provided that notice of such action shall have been given at a previous meeting.

ART. 19. Any member, associate or junior, in arrears may, at the discretion of the Council, be deprived of the receipt of publications, or stricken from the list of members, when in arrears for one year. Such person may be restored to membership by the Council on payment of all arrears, or by re-election after an interval of three years.

OFFICERS.

ART. 20. The affairs of the Society shall be managed by a Council, consisting of a President, six Vice-Presidents, nine Managers, and a Treasurer, who shall also be the Trustees of the Society.

All past (ex) Presidents of the Society, while they retain their membership therein, shall be known as Honorary Councillors, and shall be entitled to receive notices of all meetings of the Council and may take part in any of its deliberations; they shall be entitled to vote upon all questions except such as affect the legal rights or obligations of the Society or its members.

ART. 21. The members of the Council shall be elected from among the members and associates of the Society at the annual meetings, and shall hold office as follows:

The President and the Treasurer for one year; and no person shall be eligible for immediate re-election as President who shall have held that office for two consecutive years; the Vice-Presidents for two years, and the Managers for three years; and no Vice-President or Manager shall be eligible for immediate re-election to the same office at the expiration of the term for which he was elected.

ART. 22. A Secretary, who shall be a member of the Society, shall be appointed for one year by a majority of the members of the Council at its first meeting after the annual election, or as soon thereafter as the votes of a majority of the members of the Council can be secured for a candidate. The Secretary may be removed by a vote of twelve members of the Council, at any time after one month's notice has been given him by a majority of its members to show cause why he should not be removed, and he has been heard to that effect. The Secretary may take part in any of the deliberations of the Council, but shall not

have a vote therein. His salary shall be fixed for the time he is appointed by a majority vote of the Council.

ART. 23. At each annual meeting, a President, three Vice-Presidents, three Managers and a Treasurer shall be elected, and the term of office of each shall continue until the end of the meeting at which their successors are elected.

ART. 24. The duties of all officers shall be such as usually pertain to their offices or may be delegated to them by the Council or by the Society. The Council may, in its discretion, require bonds to be given by the Treasurer.

ART. 25. The Council may, by vote of a majority of all its members, declare the place of any officer vacant, on his failure for one year, from inability or otherwise, to attend the Council meetings, or to perform the duties of his office. All such vacancies and those occurring by death or resignation shall be filled by the appointment of the Council, and any person so appointed shall hold office for the remainder of the term for which his predecessor was elected or appointed; *provided* that the said appointment shall not render him ineligible at the next annual meeting.

ART. 26. Five members of the Council shall constitute a quorum; but the Council may appoint an Executive Committee, or business may be transacted at a regularly called meeting of the Council, at which less than a quorum is present, subject to the approval of a majority of the Council, subsequently given in writing to the Secretary and recorded by him with the minutes. Absent members of the Council may vote by proxy upon subjects stated in the call for a meeting, said proxy to be deposited with the Secretary.

ART. 27. The President on assuming office shall appoint a Finance Committee and a Publication Committee and a Library Committee of five members each. The appointment of two members of each Committee shall expire at the end of each year. The Secretary shall, *ex officio*, be a member of all three committees.

ART. 28. The Finance Committee shall have power to order all ordinary or current expenditures, and shall audit all bills therefor. No bill shall be paid except upon their audit. When special appropriations are ordered by the Society, they shall not take effect until they have been referred to the Council and Finance Committee in conference.

ART. 29. It shall be the duty of the Publication Committee to receive all papers contributed, to decide which shall be published in the *Transactions*, and which shall be read in full at the meetings.

ART. 30. It shall be the duty of the Library Committee to take charge of the collection of all material for the Library of the Society, and to supervise all regulations for its use.

ELECTION OF OFFICERS.

ART. 31. At the regular meeting preceding the annual meeting a nominating committee of five members, not officers of the Society, shall be appointed, and this committee shall, at least thirty days before the annual meeting, send to the Secretary the names of nominees for the offices falling vacant under the rules. In addition to such regularly appointed committee, any other five members or associates, not in arrears, may constitute an independent nominating committee, and may present to the Secretary, at least thirty days before the annual meeting, all the names of such candidates as they may select. All the names of such independent nominees shall be placed upon the ballot list, with nothing to distinguish them from the nominees of the regular committee, and the Secretary shall at once mail the said list of names to each member and associate in the form of a letter ballot, it being understood that the assent of the nominees shall have been secured in all cases.

ART. 32. In the election of Vice-Presidents, each member and associate may cast as many votes as there are Vice-Presidents to be elected. He may give all these votes to one candidate, or distribute them among more, as he chooses. Managers shall be voted for in the same way.

ART. 33. Any member or associate entitled to vote may vote by retaining or changing the names on said list, leaving names not exceeding in number the officers to be elected, and returning the list to the Secretary—such ballot inclosed in two envelopes, the inner one to be blank and the outer one to be indorsed by the voter. No member or associate in arrears since the last annual meeting shall be allowed to vote until said arrears shall have been paid

ART. 34. The said blank envelopes shall be opened by tellers at the annual meeting, and the person who shall have received

the greatest number of votes for the several offices shall be declared elected.

MEETINGS.

ART. 35. The annual meeting of the Society shall be held on the first Thursday in November of each year, in the City of New York, unless otherwise ordered, at which a report of proceedings and an abstract of the accounts shall be furnished by the Council. The Council may change the place of the annual meeting, and shall, in that case, give timely notice to members and associates.

ART. 36. Other regular meetings of the Society shall be held in each year at such time and place as the Council may appoint. At least thirty days' notice of all meetings shall be mailed by the Secretary to members, honorary members, associates and juniors.

ART. 37. Special meetings may be called whenever the Council may see fit; and the Secretary shall call a special meeting at the written request of twenty or more members. The notices for special meetings shall state the business to be transacted, and no other shall be entertained.

ART. 38. Any member, honorary member or associate may introduce a stranger to any meeting; but the latter shall not take part in the proceedings without the consent of the meeting.

ART. 39. Every question which shall come before the Society shall be decided, unless otherwise provided by these rules, by the votes of a majority of the members and associates present, provided there is a quorum.

ART. 40. At any regular meeting of the Society thirteen or more members and associates shall constitute a quorum.

ART. 41. Unless otherwise ordered, papers shall be read in the order in which their text is received by the Secretary. Before any paper appears in the *Transactions* of the Society, a copy of the paper shall be sent to the author, and, so far as possible, a copy of the reported discussion shall be sent to every member who took part in the same, with requests that attention shall be called to any errors therein.

ART. 42. The Society shall claim no exclusive copyright in papers read at its meetings, nor in reports of discussions, except in the matter of official publication with the Society's

imprint, as its *Transactions.* The Secretary shall have sole possession of papers between the time of their acceptance by the Publication Committee and their reading, together with the drawings illustrating the same ; and at the time of such reading, or as soon thereafter as practicable, he shall cause to be printed, with the authors' consent, copies of such papers, "subject to revision," with such illustrations as are needed for the *Transactions*, for distribution to the members and for the use of technical newspapers, American and foreign, which may desire to reprint them in whole or in part. The policy of the Society in this matter shall be to give papers read before it the widest circulation possible, with the view of making the work of the Society known, encouraging mechanical progress, and extending the professional reputation of its members.

ART. 43. The author of each paper read before the Society shall be entitled to twelve copies, if printed, for his own use, and all members shall have the right to order any number of reprints of papers at a cost to cover paper and printing; *provided*, that said copies are not intended for sale.

ART. 44. The Society is not, as a body, responsible for the statements of fact or opinion advanced in papers or discussions, at its meetings; and it is understood that papers and discussions should not include matters relating to politics or purely to trade.

AMENDMENTS.

ART. 45. These rules may be amended, at any annual meeting, by a two-thirds vote of the members present; *provided*, that written notice of the proposed amendment shall have been given at a previous meeting.

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PAPERS

OF THE

NEW YORK MEETING,

(XXVIIIth)

DECEMBER, 1893,

BEING ALSO THE FOURTEENTH ANNUAL MEETING OF THE SOCIETY.

DLVI.

PROCEEDINGS

OF THE

NEW YORK MEETING

(XXVIIIth),

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

December 4th to December 8th, 1893.

THE opening session, which was convened at half-past eight in the evening of Monday, December 4th, in the cozy auditorium of the house of the Mechanical Engineers' Library Association, met to listen to the address of the Hon. Eckley B. Coxe, President of the Society. His subject was "Small Anthracite as Fuel," and was illustrated by lantern slides projected upon the screen, and by models of apparatus which he had used in his investigations.

At the close of this address a collation was served in the supper-room in the lower hall, and the remainder of the evening was given to a social reunion of the members and their ladies.

Before the session adjourned, the members were invited by the President of the Engineers' Club to avail themselves of the courtesies of that institution.

Messrs. John H. Webster and Charles W. Barnaby were appointed by the President to act as tellers, under Article 35 of the Rules, to count the ballot for officers.

SECOND DAY. TUESDAY, DECEMBER 5TH.

The first session for business was convened in the Auditorium of the Society's house at 10 A.M. The secretary's register in the

reception room showed the following members in attendance during the Convention:

Alberger, L. R.,
Aller, A.,
Almiral, J. A.,
Almond, T. R.,
Almy, Darwin,
Archer, E. A.,
Ashworth, Daniel.

Baker, C. F.,
Baker, Charles W.,
Baldwin, F. R.,
Baldwin, Oscar H.,
Baldwin, S. W.,
Ball, F. H.,
Bang, H. A.,
Barnaby, C. W.,
Barr, H. P.,
Barr, J. H.,
Bates, E. P.,
Bayles, R. N.,
Beach, Charles S.,
Beach, Giles,
Beaman, E. A.,
Beardsley, Arthur,
Benham, E. B.,
Billings, C. E.,
Bonner, W. T.,
Bowen, E. S.,
Bradley, W. H.,
Bristol, W. H.,
Brown, Alexander T.,
Bulkley, H. W.,
Burden, J. A.,
Butcher, J. J.

Cadwell, W. D.,
Camp, G. E.,
Campbell, Gordon,
Canfield, Hobart,
Capen, T. W.,
Carpenter, R. C.,
Cartwright, Robert,
Cary, A. A.,
Cassier, Louis,
Chase, H. S.,
Cheney, W. L.,
Christensen, A. C.,
Christiansen Alfred,
Christie, W. W.,
Clarke, Samuel J.,
Clarke, T. C.,
Cogswell, W. B.
Conover, E. K.,
Connell, J. A.,
Conrad, H. V.,
Corbett, C. H.,
Coxe, E. B.,
Creelman, W. J.,
Cremer, J. M.,
Cronise, E. S.,
Cruikshank, Barton,
Cullingworth, G. R.

Darling, E. A.,
Davis, L. K.,
Davis, J. H.,
Dean, F. W.,
Dinkel, George,
Dixon, W. F.,
Dock, Herman,
Dodge, J. M.,
Doran, W. S.,
Durfee, W. F.,
Dutton, C. S.

Eberhardt, F. L'H.,
Edwards, L. T.,
Egleston, Thomas,
Elson, L. E.,
Emery, Charles E.,
Engel, L. G.

Faber, Du'Faur A.,
Firth, W. E.,
Fish, C. H.,
Fladd, F. C.,
Flagg, E. G.,
Fletcher, A.,
Fletcher, W. H.,
Fritz, John,
Fry, C. A.

Gantt, H. L.,
Gause, F. T.,

Gilkerson, J. A.,
Gilmore, R. J.,
Gobeille, J. L.,
Goetz, V. G.,
Gould, W. V.,
Granger, A. S.,
Greenleaf, G. E.
Gregory, William,
Grimm, Paul H.

Hallock, J. K.,
Hamilton, J. V.,
Hart, F. L.,
Hartness, James,
Henderson, Alexander,
Henderson, J. C.,
Henning, G. C.,
Herman, Ludwig,
Hibbard, H. D.,
Hill, George,
Hillmann, G.,
Hoadley, J. N.,
Hoffecker, W. L.,
Holloway, J. F.,
Hopton, W. E.,
Hough, D. L.,
Howell, E. I. H.,
Hunt, C. W.,
Huson, W. S.,
Hutton, F. R.,
Hyde, C. E.

Idell, F. E.

Jarecki, A.,
Jenks, W. H.,
Jones, H. P.

Kafer, J. C.,
Kent, William,
Kerr, Walter C.,
Kirchhoff, Charles,
Klein, J. S.,
Knight, A. F.

Laforge, F. H.,
Larger, C. A.,
Larkin, F. N.,
Leavitt, E. D.,
Le Van, W. B.,
Lewis, Wilfred,

Lockwood, E. H.,
Longnecher, C. K.,
Loring, C. H.,
Low, F. R.,
Ludlow, W. O.,
Lyall, William L.

McBride, James,
McDuffie, C. D.,
McElroy, S.,
Manning, C. H.,
Mansfield, A. K.,
Mason, W. B.,
Matton, F. V.,
Meatz, J. T.,
Mellin, C. J.,
Melvin, D. N.,
Metcalf, William,
Meyer, J. G. A.,
Miller, Alexander,
Miller, F. J.,
Miller, H. B.,
Miller, L. B.,
Miller, Spencer,
Mitchell, A. E.,
Moeller, Franklin,
Monaghan, W. F.,
Montgomery, H. M.,
Moore, L. B.,
Morison, George S.,
Morse, C. M.,
Moulthrop, Leslie,
Müller, T. H.,
Mumford, E. H.

Nason, C. W.,
Naylor, E. W.,
Nordland, Lawrence,
Norris, H. McCoy,
Norris, J. H.

Odell, W. H.

Painter, William,
Parsons, F. W.,
Pearson, W. A., jr.,
Peek, George M.,
Peirce, W. M.,
Pentz, A. D.,
Pitman, S. M.,
Platt, G. H.,

Platt, John,
Platt, J. C.,
Pollock, Alexander,
Porter, William,
Pusey, C. W.

Quimby, W. E.

Rankin, T. L.,
Raque, P. E.,
Raynal, A. H.,
Reber, L. E.,
Redwood, I. I.,
Richards, F. H.,
Richards, Frank,
Richmond, George,
Rickson, C. E.,
Roberts, Percival, jr.,
Roberts, William,
Robinson, J. M.,
Rockwood, George I.,
Roelker, H. B.,
Rogers, W. S.,
Ross, E. L.,
Rowland, A. E.,
Rowland, C. B.,
Royce, H. A.

Sanguinetti, Percy,
Sargent, C. E.,
Sargent, J. W.,
Satherberg, C. N.,
Scheffler, F. A.,
Schutte, Louis,
Seaver, J. W.,
Sewall, M. W.,
Seymour, J. A.,
Shelmire, W. H., jr.,
Sinclair, G. M.,
Smith, Charles F.,
Smith, H. W.,
Smith, G. S.,
Smith, Oberlin,
Sorge, A. J.,
Spaulding, H. C.,
Spies, Albert,
Stangland, B. F.,
Stetson, George R.,
Stiles, N. C.,
Stillman, F. H.,
Stratton, E. Platt,
Strong, George S.,
Suplee, H. H.,
Swasy, Ambrose.

Taylor, F. W.,
Taylor, J. T.,
Taylor, Stevenson,
Thomson, John,
Thurston, R. H.,
Tilden, James A.,
Torrey, H. G.,
Towne, H. R.,
Trautwine, A. P.,
Tremaine, E. G.

Vail, J. H.

Wagner, J. R.,
Walworth, A. C.,
Ward, W. E.,
Warren, B. H.,
Watson, William,
Webb, J. B.,
Webster, J. H.,
Webster, William R.,
Weeks, George W.,
Weickel, Henry,
Wellman, S. T.,
West, W. Frank,
Wheeler, F. M.,
Whitehead, George E.,
Wilcox, Charles H.,
Wiley, William H.,
Winship, J. G.,
Wolff, A. R.,
Wood, De Volson,
Wood, Walter,
Woodbury, C. J. H.,
Woolson, O. C.,
Wright, J. Q.,
Wyman, H. W.

Vanderhoef, George N.

Zehnder, Charles H.

The first business of the morning was the presentation of the Annual Report of the Council, which was read as follows:

The Council of the American Society of Mechanical Engineers begs leave to report to the membership of the Society, under Article 35 of the Rules, the following business transacted during the Society year ending November 15, 1893.

Six meetings have been held for business, at which the passing upon applications for membership and other matters came up for consideration.

Applications have been received from a number of technical institutions, requesting that their libraries might be put upon the list of those who are to receive the volumes of *Transactions* gratuitously, as they are issued each year. These requests have been granted in the cases of institutions which by their charter give to their students a degree in engineering, and, in some cases, to make their series complete, back volumes have been transmitted to such libraries at the usual members' rate of half the price at which they can be purchased by outsiders.

The Council would report the consummation of the work of its Committee, whereby, in conjunction with the other engineering societies, a headquarters for engineers was opened and maintained in the city of Chicago during the continuance of the Columbian Exposition. These headquarters were at No. 10 Van Buren Street, conveniently accessible to the down-town hotels, and included two large rooms upon the upper floor of the building, accessible to members, and the necessary additional space for the offices of the General Secretary and his assistants. Mr. Max E. Schmidt, of Chicago, was the choice of the Committee for this position. Besides these rooms in the heart of the city, by the courtesy of F. J. von Skiff, headquarters for engineers were also maintained at the Exposition grounds, in the gallery of the Building of Mines and Mining. For the courtesy of the extension of the use of this headquarters to the members of this Society, the Council is indebted to the Council of the American Institute of Mining Engineers.

To meet the expenses connected with the maintenance of the general headquarters of the associated engineering societies a subscription was collected from interested members, amounting to $2,442.45, from which the sum of $2,000, required by the Executive Committee of the engineering societies, was sent to their treasurer as requested, and the remainder will be turned

in for the general purposes of this Society, to be invested hereafter.

A Committee of the Council was also appointed, consisting of Messrs. Holloway, Baldwin, and Towne, of New York, to co-operate with Messrs. Hunt and Porter, of Chicago, for the purpose of attending to such extra matters as presented themselves in connection with the visit of foreign engineers to America during the Columbian year. For this service the above Committee decided upon the maintenance in the Society's house of a bureau of information and headquarters for the use of foreign visitors. Circulars of invitation were sent in English, French, and German to representatives of foreign societies, with the request that they might be distributed among the members of those societies intending to visit the United States. Cards were inclosed in these invitations which would accredit the visiting members to the courtesies of the Society. The seats in the auditorium were removed, the floor was stained and covered with a rug, and tables and movable chairs were supplied for the use of those who might come. The Committee thought that most valuable service might be rendered to foreign engineers if they found upon their arrival a collection of trade catalogues and other information which might prove useful to them upon the subjects they might wish to investigate during their stay, and, to this end, circulars were sent to those members interested, to which a very full response was received, and a most valuable collection of miscellaneous industrial information was gathered together upon the tables, which proved of immense service and interest; so valuable, indeed, that a movement to maintain it as a feature of the Society's house, for the benefit of our own members, is under consideration.

To meet and converse with those unfamiliar with the English language, a linguist was employed as clerk in charge of the bureau from the 1st of June to the 30th of September. The number of visitors to the house has not been as great as was anticipated, but upwards of two hundred such guests were received and entertained, all of whom gave most enthusiastic recognition of what was done for them by their American hosts.

Letters of thanks have been received from the representatives of foreign societies, expressing their indebtedness to this Society for the maintenance of the headquarters both in this city and in Chicago.

The attractiveness of the auditorium in its changed dress has so commended itself to all who have visited it, that the Council has directed that the previous arrangement of permanent seats be replaced by the more flexible and homelike arrangement which the members now see.

The Council would further report the consummation of all the business connected with the conduct of the Mechanical Section of the Engineering Congress; the publication of all the papers of that section has been completed, and the membership will receive notice of the procedure by means of which the papers read before the other sections can be procured. The unusual size of the fourteenth volume of *Transactions*, which contains the proceedings of that Congress, explains the slight delay in its issue, which has been further retarded by the necessity for the interchange of matter for that volume between this country and Europe.

The Council has also received from the representatives of its own especially appointed Committee the report of the procedure undertaken for the return of the courtesies which were shown to the representatives of this Society on the occasion of their visit to Europe in 1889, and of those who were able at that time to accept the invitation of the Institution of Civil Engineers of Great Britain and the Civil Engineers of France. The text of the report of this Committee is also made an appendix to the report of the Council.

Pursuant to action at the last annual meeting, the Council appointed a Committee on Standard Thickness for Uncoated Metal in Sheets and Plates, consisting of Messrs. Towne, Egleston, S. S. Wheeler, Oberlin Smith, S. T. Wellman, and Geo. M. Bond.

Gifts to the Society have been received during the year, of a model of the Ericsson Monitor, presented by Thos. F. Rowland, and a model of the steam yacht *Reverie*, with drawings of its fittings and construction, from Stephen Wilcox and G. H. Hillman. A portrait in oil of Joseph Harrison, builder of some of the earliest locomotives, and identified with much early American engineering, as well as with the boiler which bears his name, has been received from his widow, through the kindly interest of Mr. H. H. Suplee, his nephew. It now hangs upon the wall of the auditorium, and suitable letters of thanks have been forwarded to the donor.

Especial interest attaches to the loan, which has received the central place of honor in the auditorium, of an oil portrait of Robert Fulton, the first successful practitioner of steam navigation in this country. This has been loaned to the Society by its owner, Mr. Alanson Cary, and the particular feature of interest is that it is said to be the work of Mr. Fulton's own hands.

An interesting bust of James Watt has also been added to the Society's collection, through the kindly interest of Mr. Erwin Graves, member of the Society.

The Council has been gratified to recognize the honor conferred upon American mechanical engineering and upon this Society in the selection of its member and former vice-president, Mr. John Fritz, by the Iron and Steel Institute of Great Britain, to be the recipient of the Bessemer medal this year, and in his election to honorary membership in that institute. In an appendix to this report is the minute which was forwarded by action of the Council to the Iron and Steel Institute.

By action of the Mechanical Section of the Engineering Congress, convened at Chicago in connection with the Columbian Exposition, certain matters were referred to the Council.

It has been deemed inexpedient to comply with the request from this meeting that this Council appoint a Committee to co-operate with the Committee of the American Railway Master Mechanics' Association, for the purpose of carrying out a test at the laboratory of Purdue University of one simple and one compound locomotive.

It was the sense of the Council that while scientific investigation of fundamental data and principles would be an important and useful function of the Society, it was, on the other hand, very unwise for the Society to enter upon investigations by an appointed Committee upon issues which could be made matters of controversy by parties commercially interested, and which would involve the Society, through its representatives, in an apparent alliance with one or another manufacturing enterprise.

The Council desires to express its thanks and recognition to Mr. G. C. Henning, member of the Society and Secretary of its Committee on Standard Methods of Test, for the services rendered to the Society as delegate to the International Conference on this subject at Vienna. Mr. Henning has attended this Conference without charge to the Society for his time, and at but a moderate outlay for his expenses.

The Council has presented for action by the membership the amendment to Article 18 of the Rules, of which notice was given under the Rules at the last annual meeting.

At the end of the present Article 18 add as follows:

"And the Council shall, at its discretion, have power by a unanimous vote at a regular meeting to remit the fee for Life Membership in whole or in part, in the case of any person who shall have been not less than ten years a full member, or an associate in good standing."

The new article will therefore read as follows:

"ART. XVIII. The initiation fees of members and associates shall be $25, and their annual dues shall be $15, payable in advance. The initiation fee of juniors shall be $15, and their annual dues $10, payable in advance. A junior, being promoted to full membership, shall pay an additional initiation fee of $10. Any member or associate may become, by the payment of $200 at any one time, a life member or associate, and shall not be liable thereafter to annual dues, and the Council shall, at its discretion, have power by a unanimous vote at a regular meeting to remit the fee for Life Membership in whole or in part, in the case of any person who shall have been not less than ten years a full member, or an associate in good standing."

There has been favorable action by the voting membership during the year upon 117 applications for membership in the several grades.

The losses by death have been as follows:

James Morgan,	J. F. Haskins,	Jas. H. Fitts,
W. J. Root,	Moses G. Farmer,	Edward S. Moffatt,
W. H. Patton,	C. B. Cottrell.	E. V. Clemens,
J. F. Webster,	J. C. R. Gemmell,	Arthur F. Wendt,
A. T. Woods,	Robert Whitehill,	W. T. Nicholson,
Stephen Wilcox,	Johann Bauschinger,	J. I. Veeder.

The present membership, including those elected at this meeting and favorably acted upon by the voting membership, is 1,650, and is distributed among the grades as follows:

Honorary members	17
Life members	54
Members	1,317
Associates	77
Juniors	239
	1,650

The Council would also present the report of its Tellers to count ballots for members elected since the last meeting:

REPORT OF THE TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council, to act as Tellers (under Rule 13), to scrutinize and count the ballots cast for and against the candidates proposed for membership in the American Society of Mechanical Engineers, and seeking election before the Twenty-eighth Meeting, New York, 1893.

They have met upon the designated days, in the office of the Society, and have proceeded to discharge their duty. They would certify, for formal insertion in the records of the Society, to the persons whose names appear on the appended list, to their respective grades.

There were 598 votes cast of the blue ballot, of which 27 were thrown out because of informalities (the members voting having neglected to indorse the sealed envelope).

WILLIAM H. WILEY,
C. W. HUNT, } *Tellers of Election.*

AS MEMBERS.

Crawford, Robert,
Estrada, Esteban Duque,
Keep, William J.,
Kirk, William Fell,
Klein, John S.,
Mason, William B.,
Melville, George W.,
McFarland, Walter A.,
Nordberg, Bruno Victor,
Parshall, Horace Field,
Purinton, Arthur James,
Satherberg, Carl Hugo,
Waitt, Arthur Manning,
Wheeler, Fred. B.,
Wilmore, John Jenkins.

AS ASSOCIATES.

Baker, George H.,
Frevert, H. F.,
Garfield, Lyman Morse,
Prosser, Thomas, Jr.

FOR PROMOTION TO FULL MEMBERSHIP.

Foran, George Jesse,
Grist, James Edmund,
Torrance, Kenneth,
Wiggin, William H.

AS JUNIORS.

Beals, Albert E.,
Carpenter, Henry Albert,
Goetz, Victor J.,
Hoxie, George L.,
Johnston, William A.,
Jones, Richard Oscar,
Marble, Harry Mills,
Piers, Frank,
Reid, Joseph Snively,
Russell, Edmund Allen,
Rutherford, Walter M.,
Stanley, Alexander Welch,
Van Trump, C. R.

Respectfully submitted by

THE COUNCIL.

APPENDIX I TO REPORT OF COUNCIL.

MINUTE TO THE IRON AND STEEL INSTITUTE OF GREAT BRITAIN.

It having come to the knowledge of the Council of The American Society of Mechanical Engineers that the Iron and Steel Institute of Great Britain has recently bestowed upon a member and ex-vice-president of the Society the "Bessemer Medal" for 1893, the Council avails itself of this occasion to tender its congratulations to Mr. John Fritz on the receipt of such a well-merited and distinguished honor, from his friends beyond the sea.

The Council desires to commend the excellent judgment of the Iron and Steel Institute in selecting for this honor one who, by his genius, skill, and industry through many years, has done so much toward the improved metallurgical processes, and especially in the construction of the mechanical appliances now in use in the manufacture of iron and steel in this country, as to have materially aided in making this industry the important one it now is.

That in conferring the "Bessemer Medal" in this the year of our "Columbian Exposition" upon an American engineer, the Institute has in a pleasant way added further evidence of the kindly feelings and hearty good-fellowship which in so many ways has of late marked the action of engineering societies and engineers in this country and in Europe, and we cannot but believe courtesies like this tend to a closer union of engineers in all lands; that while it is the mission of our profession to compel the rude forces and materials of nature to contribute to the welfare and comfort of mankind, it is their mission as well to bring in closer touch nations and individuals, and in this way, aiding in a degree it would seem no other profession can, to bring about those better days which in all ages past the world has hoped and struggled for.

APPENDIX II.

SUMMARY OF REPORT CONCERNING THE ENTERTAINMENT BY AMERICAN ENGINEERS OF THEIR HOSTS OF 1889.

AT a meeting of the Council, held in New York city, August 12, 1891, a Committee was appointed to consider the question of the procedure to invite the English and French engineers to become the guests of the American engineers during the World's Fair.

This Committee reported at the meeting, October 15, 1891, a plan for the scope of such entertainment, and a letter was addressed to the sister engineering societies. The first meeting for conference was held April 18, 1892. Meanwhile invitations had been sent to the engineers of all Europe to visit America for the Congress of Engineering, and it was the opinion of the conference meeting that two distinct obligations were laid upon the profession. The first was an entertainment by the American societies of all European engineers, and the second was an entertainment by those engineers participating in the excursion of 1889, of those who were then their hosts. For these separate functions two separate committees were appointed, by action of the Council, July 2, 1892.

The joint Committee appointed under this resolution held several meetings during the spring and autumn of 1892, and on January 26, 1893, issued a circular to the members of the party of 1889, in which, in the name of the four national engineering societies, they requested subscriptions for the purpose of the return of the courtesies of that year. Subscriptions were received from members of this party, aggregating a sum in the neighborhood of $7,000, of which a portion was at once paid in. Correspondence was therefore commenced with the Secretary of the Institution of Civil Engineers of Great Britain, and the Society of Civil Engineers of France. It became manifest, as the spring advanced, that the exertions of Mr. Forrest, Mr. Dredge, and Sir Benjamin Baker (the two latter honorary members of this Society), by personal letter and by circular of inquiry, would not result in the gathering together of a party of British engineers of any size to visit America in a body; strenuous exertions to

bring this result about on the part of the Committee proved fruitless, and a plan of an entertainment to include the British engineers had to be reluctantly abandoned.

The Marquis of Chasseloup-Laubat, a member of the French Commission to the Columbian Exposition, and also a member of the French Society of Civil Engineers, had made definite arrangements for the visit of the French engineers, pursuant to the invitation of the Committee, and had secured a party forty-six in number, of gentlemen alone, for the trip to the United States. The entertainment given in the name of the Eighty-niners, therefore, expended itself upon the courtesies shown this party of French engineers. A brief outline of what was done embraces the following programme: *

GENERAL PROGRAMME OF THE VISIT OF THE FRENCH ENGINEERS TO AMERICA AS GUESTS OF THE EIGHTY-NINERS.

Saturday, August 26 ..... Leave Havre on *La Champagne.*
Sunday, September 3 ..... Arrive at Morton Street Pier, N. Y.
September 3–September 7 ..... Guests of New York Welcome Committee.
Monday ..... Bridges and Park Drive.
Tuesday ..... The city.
Wednesday ..... The river and harbor.
Thursday ..... The respite for business.
Friday, September 8 ..... Special Complimentary Train to Niagara.
Saturday, September 9 ..... At Niagara. Leave by Special Train for Chicago.
September 10–19 ..... In Chicago. (Auditorium.)
Tuesday, September 19 ..... Leave for St. Louis.
Wednesday, September 20 ..... In St. Louis and leave for Pittsburgh.
September 22, 23 ..... In Pittsburgh. (Monongahela.)
Sunday, September 24 ..... *En route* for Washington, D. C.
Monday, September 25 ..... In Washington. (Ebbitt.)
September 26, 27 ..... In Philadelphia. (Continental.)
Thursday, September 28 ..... Arrive New York.
Friday, September 29 ..... Final day in New York city.
Saturday, Sept. 30, to Sunday, Oct. 8 ... *En route* for Havre.

Of this programme the Committee assumed the burden of entertainment at Niagara and at Detroit, and also bore a portion of the entertainment of a small group of the guests who made a side trip into Canada after the departure of the main body for home.

* Appended to the original report was a list of the party, badge, sample programme, etc., etc.

The fact that the itinerary of the French engineers was arranged by a representative of theirs, rather than by the representatives of the Committee alone, made necessary a change in the position of the Committee from that which they had taken in their original circular. Following the precedent of 1889, it was the plan of the Committee to show attention to the party on their way from the seacoast to the Exposition, and to leave them to return eastward as they might find convenient. The policy of keeping the entire party together during the entire round trip made it necessary that the cities visited on the return should also bear a portion of the burden of entertainment. The Committee therefore directed that the subscriptions received from residents in or near Pittsburgh, Chicago, St. Louis, Washington, and Philadelphia should be returned to the subscribers, to enable them to meet this call. There was therefore returned to subscribers, either in cash or in promises, as follows:

Chicago	$755 00
Philadelphia	670 00
Pittsburgh	250 00
St. Louis	150 00
Washington	25 00
Total	$1,850 00

The actual outlay of the entertainment at the East and to the West was, therefore, borne by New York city and state, and by New England.

The entertainment in New York city consisted of a luncheon at the Engineers' Club, a drive through the parks, a visit to the Elevated Railroad system, to Brooklyn Bridge, and the Grand Central Station terminal, with a lunch at the Café Savarin; and on the third day a trip around the harbor, with a luncheon, on the Fall River Company's steamer *Puritan*. The party was then dispatched by special train, complimentary from the New York Central and Hudson River Railroad (except as to the dining cars), to Niagara, thence, *via* Detroit, to Chicago, when the responsibility of this Committee was held to cease, save so far as the minor entertainment at Boston was concerned, upon the completion of the main trip. The entertainment commenced September 3, and ceased September 10. The party returned to France September 30.

The report of the receipts and disbursements of the Committee is as follows:

Total amount cash received from subscribers		$5,114 50
" cost, New York entertainment.....	$1,282 52	
" cost, Niagara, Detroit, and Chicago	788 66	
" cost, Boston entertainment (borne by this fund).....	98 90	
" cost, general expenses.....	266 36	
" cash subscriptions paid in and refunded to subscribers	270 00	
" cash on hand to balance	2,408 06	
		$5,114 50

This cash balance will be refunded to the subscribers in the form of a dividend in proportion to their subscription of 48%, to which has been added an extra dividend of two dollars apiece to each subscriber.

There remain a few uncollected subscriptions to the fund, amounting to a sum between five and six hundred dollars, from twelve members of the party, and from many of these letters have been received requesting that they may be excused from redeeming their subscription, as it is not needed; and, therefore, as this sum is less than 10% of the total, the Committee does not expect to press for its payment.

The Committee would request, therefore, that with the completion of the duty committed to it, said Committee be discharged.

APPENDIX III.

FROM THE COUNCIL OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS TO THE MEMBERS OF SISTER SOCIETIES OF ENGINEERS IN EUROPE, SOUTH AMERICA, AND THE COLONIES.

NEW YORK CITY, U. S. A., *April*, 1893.

Dear Sir,—The President and Council of the American Society of Mechanical Engineers, supplementing the general invitation already extended to foreign engineers visiting America in 1893, by the four national societies of American engineers, hereby invite you, as a representative of the engineering profession, to avail yourself of the hospitalities of its house in New York during your stay in this city.

During the season of the Columbian Exposition the Society will maintain in its house a bureau of information for its foreign guests, connected with which will be a linguist familiar with foreign languages, and thus better qualified to assist the visitors unfamiliar with either the language or customs of this country. It is proposed also to furnish visitors with information concerning places of engineering interest which they may desire to visit, and concerning also the best methods and the approximate expense of reaching them.

You are further invited to make use of the house of this Society as a place for professional or business appointments, and to use it, if desired, as an address for letters and telegrams, although, as a rule, the more customary practice of utilizing the services of your banker for this purpose is preferable.

As a representative of one of the principal branches of American engineering, this Society will always extend a cordial welcome to visiting engineers from abroad, and desires especially to promote the convenience and comfort of those who may visit the United States during the World's Columbian Exposition in 1893, among whom we hope to learn that you are included.

On behalf of the President and Council,

Very respectfully,

F. R. HUTTON, *Secretary.*

At the close of the Report of the Council the second order of business was the report of the Finance Committee, which was as follows:

The Finance Committee of the American Society of Mechanical Engineers would respectfully report to the Council the following statement of Receipts and Disbursements on behalf of the Society, under their direction, during the year from November 1, 1892, to November 1, 1893:

ANNUAL REPORT.

RECEIPTS.

Initiation Fees	$3,190 00
Current Dues	20,141 77
Past Dues	751 70
Advance Dues	242 10
Sales of Publications	1,447 46
Binding	20 00
Rent, Parlor and Hall	410 00
Badges	640 25
Engraving	82 61
Life Membership (Bonds Surrendered)	2,100 00
" " (Cash)	900 00
Contingencies	35
Postage and Express	11 05
Interest on Investment	512 50
Gift (Bonds)	900 00
House Supplies and Furniture	145 34
Chicago Headquarters	2,442 45
Certificates	50
Office Expenses	2 05
Cash in Treasurer's hands Nov. 1, 1892, as per last report	1,628 44
Interest on Balance in Savings Bank	95 26
Total	$35,663 83

DISBURSEMENTS.

General Printing and Stationery		$1,783 83
Reprints and Publications		4,684 45
Postage and Express		1,610 83
Salaries		6,101 00
Office Expenses		578 79
Engraving		1,366 60
Contingencies		119 40
Binding		1,394 00
Meetings		598 43
Work of Committees		355 73
House Supplies and Furniture		945 16
Badges and Certificates		649 70
Traveling		250 00
Insurance and Safe Deposit		44 00
Rent, Interest, and Taxes		2,297 90
Investment, Bonds rec. as L. M.		2,100 00
" " " " Gift		900 00
Interest Savings Bank Deposit		95 26
Library		273 07
Headquarters, Chicago		2,000 00
" 12 West 31st Street, New York City		156 00
On hand to balance. Bonds bought and held as an investment	$7,200 00	7,359 68
Cash in Treasurer's hands Nov. 1, 1893	159 68	
Total		$35,663 83

In explanation of and comment on the above report, the Committee begs to call attention to the fact that this is the close of the second year since an increase of $5 per annum was made in the dues of all members. It will be noticed that the total received from dues, sales of publications, etc., is $27,131.38, and by summing up all the expenses of running the Society they are found to reach a total of $23,019.90, showing that the income of the Society from dues and other sources, exclusive of initiation fees and life memberships, is in excess of the running expenses by $4,111.48, leaving receipts from these latter sources to be invested as capital from which the Society can derive an income.

The surplus this year, amounting to $7,232.90, has been devoted to the purchase of bonds issued by the Mechanical Engineers' Library Association for the purchase of real estate owned by that association. and it is the use of the surplus in this way which explains why the cash in bank has been drawn

down to the comparatively small sum which appears in the report. These bonds bear interest at 5%, and are held by the Council as an investment.

There have also been received in exchange for life membership similar bonds to the amount of $2,100, and nine bonds to the amount of $100 each have been presented to the Society by the following gentlemen: Sumner Hollingsworth, $600, and Henry R. Towne, $300. This makes, with the $5,700 reported as belonging to the Council a year ago, a total of $16,100 worth of bonds held by the Society as an invested fund, this being one-half of the entire amount of the bonds issued in the purchase of the Society house.

There has also been received, in response to a special call for subscriptions, the amount of $2,442.45 for the expenses of the Chicago headquarters for use of the members of this Society and their professional guests during the Columbian Exposition. The quota requested from this Society was $2,000, which sum has been paid to the Treasurer of the Committee of the Associated Engineering Societies, and the balance of the fund remains in the possession of the Society.

There also stands to the credit of the Society, in the Bleecker Street Savings Bank, $1,293.85, and in the Merchants' Clerks' Savings Bank of this city the sum of $1,160.38, making a fund of $2,454.23 to be devoted to the purchase of books for the library.

The expenses under the Work of Committees are this year heavier than usual, by reason of the appropriation made at the last annual meeting for the expense of a delegate to the International Congress at Vienna on Unification of Methods of Testing Materials. The expense on this account borne by the Society was $238.91, the balance of the sum necessary ($72.09) being met from a fund contributed some years ago for the use of this Committee. The other items of expense in this account were traveling and other expenses of the Committee to report a Standard Method of Testing Locomotives.

There remain outstanding against the Society for the year which has just closed an unpaid bill of Messrs. J. J. Little & Co., for printing the volume of the *Transactions* which has not as yet been distributed, amounting to $2,332.10, for the payment of which there has not been sufficient money in the treasury at any one time, since the business depression has made collection diffi-

cult. There is also outstanding the sum of $750, due the Mechanical Engineers' Library Association for rent. It will be readily understood that the Finance Committee has found collections difficult since last July, and the existence of this indebtedness, concurrent with a surplus of receipts over expenditures, as shown by the statement, is explained by the fact that the large purchase of bonds was effected before the business depression.

The uncollected dues for the year just ended also, for the same reason, are slightly larger than usual, and amount to $1,823.23 from 159 members. There is every probability of all this amount being collectible, and the indebtedness referred to will be wiped out as soon as the bills for the current year fall in.

Respectfully submitted by

THE FINANCE COMMITTEE.

REPORT OF LIBRARY AND HOUSE COMMITTEE.

The House Committee of the Society, intrusted with the duties of superintending the interests of the library, which is maintained by the Mechanical Engineers' Association as part of the consideration which passes between the two organizations in their relations as landlord and tenant, beg leave to report:

That the use of the library by students and other readers, as well as by the members of the Society, has steadily increased. A register of those who make use of it shows that its privileges have been enjoyed by over two thousand persons during the year, particularly in the evening and upon holidays. The Committee makes a special point of keeping the library open at these times, when the absence of demands of other business make it possible for those closely engaged to avail themselves of the privileges extended.

During the winter of 1893 a number of members arranged to assemble on stated evenings for profit and entertainment, the evenings closing with a social reunion. The topics of the 1893 evenings were "The Geography of the Moon," the "Buildings of the Columbian and other Expositions," "The Orchestral Phonograph," and one evening of music and recitations. These reunions were in no sense meetings of the Society, and their expenses were entirely borne by those in attendance and who looked after their details.

A movement is now on foot to carry out in a similar informal

manner a series of meetings at which some matter of professional interest shall be a topic of discussion. It should be understood that both these series of reunions are individual affairs, and have no official relation to the Society as an organization.

The report of the finances of the Mechanical Library Association is made to the entire membership of the Mechanical Engineers through the channel of the House and Library Committee, and is appended to this report.

The sleeping apartments upon the upper floors of the Society's house have been abundantly used during the entire year, and in some cases the demand for this accommodation has far exceeded the supply. The plan of having apartments of this sort at the service of non-resident members has been very popular and has been warmly supported, and will be continued as a feature of the Society's life. Those who make use of these facilities find themselves enjoying all home comforts, and the opportunity for meeting their professional brethren in a way which has proved most enjoyable. The income from this source, which passes to the credit of the Library Association, will be seen to reach nearly the sum of $1,200.

LIBRARY ASSOCIATION.

ANNUAL REPORT OF THE TRUSTEES OF MECHANICAL ENGINEERS' LIBRARY ASSOCIATION, 1892–1893.

THE summary of Receipts and Disbursements of the Trustees from November 1, 1892, to November 1, 1893, is as appended below:

RECEIPTS.

Balance on hand November 1, 1892		$775 62	
Receipts.—Fellowship Fund	$414 00		
Sinking Fund	770 50		
Library Permanent	73 65		
Office Rent	3,210 00		
Room Rent	1,193 10		
Equipment	1 95		
Rent of Roof	25 51		
Total Receipts, 1892-1893		$5,688 71	
Total Cash, 1892–1893			$6,464 33

DISBURSEMENTS.

Interest on Mortgage	$1,485 00		
" " Bonds	1,595 00		
Salaries	840 00		
House Supplies, etc.	285 75		
Fuel	249 00		
Lighting, gas, $149,62; electric, $494.57	644 19*		
Equipment	451 50		
Laundry	285 00		
Repairs	119 95		
Book purchase	132 85		
Contingencies	90 55		
Total Disbursements		$6,178 79	
Cash in Bank to balance		285 54	
			$6,464 33

ASSETS.

House and Lot No 12. West 31st Street, New York City.	$65,000 00	
Furniture and Equipment	5,000 00	
Books and MSS	10,000 00	
Total	$80,000 00	
Bills receivable (Office and Room Rent), uncollected	1,214 60	
Bills receivable (Subscriptions to Fellowship Fund), uncollected	69 00	
Bills receivable (Subscriptions to Sinking Fund), uncollected	305 50	
Total Assets		$81,589 10

LIABILITIES.

First Mortgage, held by N. Y. A. of M.	$33,000 00	
Second Mortgage, held by Members of A. S. M. E.	15,900 00	
Second Mortgage, held by Council of A. S. M. E. as an investment	16,100 00	
		65,000 00
Excess of Assets over Liabilities		$16,589 10

At the close of the report of the Council, the Tellers appointed at the previous session to count the ballot for officers to be elected

* Electric light for nine months only.

by letter ballot in advance of this meeting, presented their report as follows:

December 5, 1898.

The tellers of election to count the ballots for officers of the Society for the ensuing year present the following report:

Whole number of ballots cast		581
President	Eckley B. Coxe	580
"	Scattering	0
Vice-Presidents	C. E. Billings	580
" "	Percival Roberts	580
" "	H. J. Small	581
" "	Scattering	0
Treasurer	William H. Wiley	581
"	Scattering	0
Managers	John B. Herreshoff	582
"	L. B. Miller	578
"	W. S. Russel	580
"	Scattering	0

There were twenty votes cast out by reason of informalities (the members voting having neglected to indorse the sealed envelope).

Respectfully submitted,

J. H. WEBSTER,
CHARLES W. BARNABY, } *Tellers of Election.*

The amendment to the rules presented by the Council, under the provisions for such amendments, was presented for consideration. In its original form the amendment as amended was read by the Secretary:

At the end of the present Article XVIII. add:

"And the Council shall, at its discretion, have power by a unanimous vote at a regular meeting to remit the fee for Life Membership in whole or in part, in the case of any person who shall have been not less than ten years a full member, or an associate in good standing; provided that due notice of the proposed action shall have been given to the Council at the regular meeting preceding that at which the action is to be taken."

In answer to an inquiry by one of the members as to the intention of such amendment, and the reason why it was necessary, the Secretary, at the request of the Chair, made the following explanation:

Prof. F. R. Hutton.—About a year ago one of the most honored members of the Society and a vice-President, since deceased, withdrew from the Society. The reason for that withdrawal was that, by increasing age and infirmity, he did not

feel that he could remain in active connection with the Society, and continue to pay its dues. His connection with engineering in this country was so intimate, so distinguished, and so close that the Council regretted that the necessity should arise for his severing his connection with the Society for any such reason. The question was agitated whether he should be elected to honorary membership. There seemed to be reasons why that was an undesirable precedent, and the policy of making honorary members for this reason seemed to be one which the Council hesitated to adopt. What was desired, therefore, was that there should be created for persons of this sort—and there were several—a grade which should be the equivalent of a retired list, in which members who had been for ten years or more identified with the Society—charter members of it, perhaps—should be retained upon the Society's rolls, their interest and the prestige of their names still attaching to the Society, without their being exposed to the prohibitory requirement of the payment of full yearly dues. The impossibility of election to honorary membership due to the limitations imposed in the Rules to the number on the honorary list, and the fact that this difficulty might arise in cases where election to honorary membership might not be the wise and proper expedient to meet the case, were reasons why it did not seem advisable to establish the precedent that such persons should be put on the honorary list. It is undesirable that a retired list should be created by name, and it seemed that the plan embodied in the amendment, if surrounded with the proper safeguards, was the most desirable way in which this particular difficulty might be met. The occasion which started this amendment has now passed away by the death of the honored member. There are others, however, to whom the same policy would apply. In order to surround it with safeguards, there are provisions, as the members will notice, that this can only be done by unanimous vote; that it can only be done at a regular meeting; that it can only be done in the case of those who have been at least ten years in the Society—no doubt, in time, that limit could be extended—and that due notice shall have been given of such proposed action in advance, so that none but a representative meeting of the Council shall ever give this privilege, and only where the conditions are such that it would be perfectly wise and safe to do it. That is the occasion of the amendment.

The President.—I wish to say that when I first saw this

amendment it did not strike me as a proper one. But when I heard the particulars of the case which it was intended to cover, it almost brought tears. None of us know what may happen to us when we are sixty-five or seventy, and to contemplate that a man who had stood high in the profession, has been an honored member of this Society, could be dropped from the rolls because he had a stroke of apoplexy which prevented him from earning his bread, makes us feel that there ought to be some way of covering such a case. Of course, you cannot have a list of men who cannot pay their dues.

Mr. W. S. Rogers called attention to the possibility of a misunderstanding which might follow from the use of the word "remit" in an unusual sense in the amendment as proposed, whereupon the amendment was withdrawn, to be recast into a less objectionable form, and it was presented at the next session in the following resolution:

Resolved, that there be added at the end of Article XVIII. of the Rules, the following words:

"And the Council shall have the power, by an unanimous vote, at any regular meeting, to admit to the grade of Life Member, without the payment in full, or in part, of the sum above named, such persons as for ten years or more have been either a full Member or Associate, and who, in its judgment, are entitled to this grade. Provided, that notice of such intended action shall have been given at a previous meeting."

In its amended form, the amendment was passed unanimously.

At the Chicago meeting, when the Society was in session as Section "B" of the Engineering Congress, a report of its Committee on Standard Methods of conducting Tests to determine the Efficiency of Locomotives was presented, read, and discussed. In view of the fact that this was a public meeting at Chicago, embracing members outside of the Society, it did not seem proper to have that meeting take action upon this report, so that such action was deferred until this annual meeting. The report was presented on behalf of the Chairman by title, and the following resolution was offered, duly seconded, and carried unanimously:

Resolved, that the Report of the Committee to recommend a Standard Method of Test of Efficiency of Locomotives be accepted and printed in the *Transactions* as the recommendation of that Committee.

Resolved, that the Committee be discharged with the thanks of the Society.

A similar procedure was followed with respect to the report of the Society's Committee on Standard Diameters of Flanges in

Valves and Pipes. It was presented on behalf of the Chairman by title, and the following resolution was offered and seconded and carried:

***Resolved*, that the report of the Committee to recommend Standard Sizes for Flanges of Pipes and Valves be accepted and printed in the *Transactions* as the recommendation of the Committee, and further, that the Committee be discharged with the thanks of the Society.**

The Committee on Uniform Standards for Test Specimens and Testing Materials presented no report, in view of the full presentation made at Chicago, and, as the work of the Committee was not completed, the Committee was continued without formal action.

At the close of this routine business the President asked if there were any other matters to be presented, and the session thereupon took up the discussion of professional papers. Mr. A. K. Mansfield presented the first paper, entitled "The Buckeye Engine Valve Gear," which received discussion by Messrs. Rockwood, Holloway, Kent, Hartness, Rogers, and Ball.

Prof. R. H. Thurston read a paper on "Maximum Contemporary Economy of the High Pressure Multiple-expansion Steam Engine," which was discussed by Messrs. Peabody, Ball, Rockwood, Scheffler, Kent, and Denton.

The Society thereupon adjourned until the evening.

Third Session. Tuesday Evening, December 5th.

The session was opened by a paper by Mr. George A. Morison on "Expansion Bearings for Bridge Superstructures," discussed by Mr. G. C. Henning.

Mr. Barton Cruikshank read a paper on a "Device for Drill Jigs," and Mr. F. A. Scheffler a paper on a "Curve Delineator," illustrated by the apparatus itself.

Prof. Thomas Egleston, Chairman of the Society's Committee on Standard Thickness Gauges, presented, on behalf of the Committee, the following report:

The Committee on Gauges respectfully report that they have carefully considered the matter referred to them, and they recommend that the thickness and diameter of pieces to be measured shall be expressed in decimals, which would imply the use of the micrometer screw gauge, or a permanent gauge derived from it. The Committee is divided in its opinion as to whether this decimal should be thousandths of an inch or hundredths of a millimeter.

Your Committee have conferred with other societies, and find that the American Institute of Mining Engineers, after considering the subject for more than a

year, in which every known gauge was examined, in the year 1877 made a unanimous report in favor of the adoption of the micrometer screw as the best gauge to be used by all manufacturers. Committees of the American Institute of Electrical Engineers, and of the Electrical Light Association, as appears in the report of the Society, at the last annual meeting made reports favorable to the same subject. The American Society of Civil Engineers have as yet, we believe, taken no action upon this subject, but expect shortly to do so. Your Committee are, however, of opinion that all the engineering and scientific societies of this country will join this Society if they adopt a decimal gauge.

At the first meeting of the Committee it seemed desirable that this matter which is of so much importance, should be made as far as possible international. Your Committee therefore decided to correspond with some of the societies in Europe, the result of which is, that both the French and German societies are found to be in favor of a decimal gauge. Both the French and German governments, at suggestions made by committees of your Society some years ago, have adopted standards, and are favorable to the adoption of an international decimal gauge. The Committee have learned incidentally that the highest authorities in England are also favorable to the same thing. Correspondence with the engineering societies abroad has failed to elicit a response from the Institution of Civil Engineers, and the Mechanical Engineers of that country are not willing to have the matter discussed in the Society unless it is brought before them officially.

Your Committee therefore recommend that the subject be recommitted to them, with instructions to correspond with the foreign societies, with a view of adopting an international decimal gauge.

The German, Austrian, French, and Belgian societies will undoubtedly recommend the millimeter as the basis. It is probable that the government officials of England, and probably some of the societies, will also recommend it, but without official action by the Society your Committee finds itself unable to act officially in the matter.

All of which is respectfully submitted.

T. EGLESTON, *Chairman*,
For the Committee.

It was thereupon moved that the recommendation of the Committee be approved and the matter be recommitted to the Committee, with power to act in pursuance of its recommendation. This was amended by motion that the matter be referred to the Council, but the amendment was voted down, and the original resolution was passed, that the report be accepted, and the Committee be continued with power.

The paper by Mr. Fred. W. Taylor, "Notes on Belting," was discussed by Messrs. Rogers, Nason, Cartwright, Lewis, Kent, and Towne.

Prof. R. C. Carpenter reported some experiments on the effect of Water Hammer, which was discussed by Professor Wood and Mr. Nason.

The session thereupon adjourned.

Fourth Session. Wednesday, December 6th.

This session was convened at 10 A. M., for professional papers, and proved to be one of the most interesting of the meetings. It was opened by a paper by Professor Carpenter, entitled "Constants for Correcting Indicator Springs that have been Calibrated Cold," and was discussed by Messrs. Henning, Coxe, Rockwood, Porter, Barnaby, Almond, and Cary.

Prof. W. S. Aldrich presented a paper of similar type on the "Use of the Indicator for Continuous Records in Dynamometric Testing," discussed by Professor Carpenter, who also read his second paper, entitled "New Form of Prony Brake."

Prof. D. S. Jacobus then read a paper, entitled "Comparison of the Mean Effective Pressures of Simultaneous Cards taken by Different Indicators," discussed by Messrs. Carpenter and Rockwood. This was followed by a topical discussion as to the best method of getting the average work of a steam-engine, operating under wide variations of load, which was discussed by Messrs. Platt, Kent, Rockwood, Webb, Durfee, Dutton, Ball, Cary, Low, Ashworth, Richmond, and Carpenter. Other topical queries were then taken up until the hour of adjournment, and covered the subjects of Deterioration of Copper in pipes for Expansion Joints, the Substitution of Pressure for Drop Forging in the Manufacture of small stamped Articles of Metal, and a specially interesting discussion on Packing for Rods and Plungers of Hydraulic Machinery.

This was participated in by Messrs. Rockwood, Graves, Coxe, Durfee, Raynal, Emery, and Kent.

The evening of Wednesday was devoted to a Social Reunion and Reception, with supper and dancing, held at Sherry's, Fifth Avenue and Thirty-seventh Street, at which three hundred persons were present, and which was much enjoyed by all participating.

Fifth Session. Thursday, December 7th.

This session opened at 10 A.M. by the presentation to the Society of a portrait of Prof. Francis Reuleaux, honorary member of the Society; the portrait being the gift of H. H. Suplee, a member, and the work of Miss Suplee, his sister. The portrait was hanging upon the wall, draped, and was unveiled at the close

of the presentation remarks. The speech of Professor Thurston, presenting the portrait on behalf of the donor, and of President Coxe in accepting it, were as follows:

Prof. R. H. Thurston.—I am requested by Mr. Suplee, a member of this Society, to present to the American Society of Mechanical Engineers a portrait of Dr. Francis Reuleaux, of Berlin, Germany. It is not necessary to say very much of the man; you will presently see it is not necessary to say very much of the portrait; and my task is very soon and very easily performed.

Dr. Reuleaux, as you all are aware, has for many years been an honorary member of our Society, and presumably our strongest representative on the continent of Europe. I suppose he is regarded by all the members of our Society, and by all his colleagues at home and abroad, as being in the front rank among the scientific representatives of our profession. If engineering is a learned profession, it has in Dr. Reuleaux one of its ablest, possibly its ablest, representative in Germany. The work which Dr. Reuleaux has done for the profession of engineering generally is well known to all. As an educator, as an author, as a scientist, as an economist, he stands among the leaders in his own, or in any, land. Dr. Reuleaux has been an educator—an educator of engineers—for now more than a generation. His work has been done mainly at Berlin, and he has built up there one of the noblest technical schools on the continent of Europe. In the course of this work he has stimulated the progress of engineering education throughout the world. He has been a leading figure in Germany. He has made his own an illustration of the best modern technical school, and an example for the emulation of leaders and organizers of schools all over Europe, and has in all these various ways shown himself one of the greatest men in his profession in our time. Those who have had occasion to assist in the building up of technical schools owe more, perhaps, to him than to any other man in Europe. All who are engaged in the work of education owe a great deal to Dr. Reuleaux for the lead he has taken, for the grand work he has done, in developing the professional side of engineering education, and in promoting every line of engineering progress.

You are all familiar with his great book, *Der Constructeur*, and with his other works written earlier and later. Those who have seen the models of kinematic movements designed by Dr. Reuleaux

will recognize in their proportions, in their classification, in the method of their arrangement, the work of a great mind—a mind of a strong logical order, the mind of a man who is by nature fitted to be an engineer.

I have said that he is something of an economist. Some of you may remember that at the Philadelphia Exposition of 1876 Dr. Reuleaux was one of the German Commissioners, and those of you who have seen or read or heard of his *Briefe aus Philadelphie* will remember that he criticised his own countrymen severely, because they had not succeeded in placing beside the exhibit of machinery in the United States section an equally good exhibit from Germany; and he fell into ill odor with many of his own countrymen because of his frank report and his honesty. But that he was, nevertheless, a great political economist is shown by the fact that, when the Germans made their appearance at Chicago last summer, they placed in every department of that great cyclopedic Exposition exhibits that compared favorably with those of any country in the world, and the stimulus given to the finer class of manufacture, to fine, conscientious, good work, by the frank letters of Dr. Reuleaux has been seen by all who have visited the Exposition this last summer.

I have known Dr. Reuleaux for now over twenty years. Twenty years ago, visiting the Exposition at Vienna, I found him engaged upon the work of the International Jury. As his colleague I saw him daily, and became daily better and better acquainted with him. I have spoken of him as an educator, as an author, as an economist, but Dr. Reuleaux, as a man, stands head and shoulders above Dr. Reuleaux as educator, as writer, as economist. He is one of the noblest of nature's noblemen, and no one can have known him without having learned to love him; and I enjoy highest gratification in the thought that I am to-day afforded the pleasure and the honor of presenting the portrait of such a man —great, good, lovable—as a contribution to the gallery of the American Society of Mechanical Engineers, and as a gift from a member of the Society.

The portrait was painted by Miss Suplee, the sister of the donor, and we are therefore indebted to the lady as well as to our fellow-member, our colleague, for this most acceptable gift. It was necessary that Miss Suplee, not being well acquainted with Dr. Reuleaux, should paint this portrait mainly from photographs, but she met her subject once at Philadelphia, and the lifelike touch

by which we always recognize a real portrait will be here seen as that given as a memory of her meeting with our noble friend.

It, therefore, gives me very great pleasure, Mr. President, to tender to you, for the Society, in behalf of Mr. Suplee, the donor, and of Miss Suplee, the painter of the portrait, this, which we consider a a worthy contribution to the collection of portraits just beginning to take form in our audience room; and we hope that this contribution will be no less welcome than those that have preceded it, and that the portraits now seen here will prove to be only the beginning of a long and splendid list that shall in future times worthily represent the talent, the genius, the honors, in every branch of engineering, that shall be contributed through its most distinguished members by our Society, to our country and to the profession, at home and abroad.

President Eckley B. Coxe.—That it is a pleasure on my part, on behalf of this Society, to receive this portrait, goes without saying. As we look around and see the portraits which are about us we realize the broad field of mechanical engineering. While their numbers are not great, they represent a very great number of lines of work and thought. For example, there we have Professor Rankine, representative of theoretical engineering, of high theory applied to practical work, a man whose work did not come into as great use in his lifetime as it did after his death, and to-day you can hardly read any paper or listen to any number of papers upon the subject of engineering where theory is discussed without hearing him cited. Above him here we have (pointing to Holley's portrait)—I need not mention his name, gentlemen, you all know it—showing what high talent, brilliant capacity did to make commercially successful the manufacture of steel and whatever else he took up in this country. There we have Sir Henry Bessemer, and I could go in this way around the room. The portrait just presented is that of another type of educator. Professor Reuleaux, as Professor Thurston has so well said, has taught the teachers; the men who have been founding the schools, the men who have been studying the question of how to train the young men, go to him. But he is also a great teacher of students. As some of you may remember, he said a few words at Chicago, and they produced an impression upon my mind. They proved the great, broad, practical views of the man, who, from his books, one might think would rather discuss theoretical questions and leave them to be practically carried out by other people. He said that

what astonished him in the Chicago Exposition most was the accuracy of measurement of the American machinist, and then he made what was a very curious remark. He said, "Your people here understand perfectly distinctly what a thousandth of an inch is." He said, "I am not talking about the professors or the experimenters, but in almost any good hardware shop in any large city you can buy gauges that measure one thousandth of an inch. Now the fact that they sell them is very decided evidence that somebody buys them, and the fact that people buy them all over the country shows that there must be a very large number of your practical mechanics who are trained to measure thus closely. I am not, of course, referring to the actual work done by them, for everywhere first-class mechanics are to be found, but only to their knowledge of accurate measurement." That showed that the man not only was strong in the theoretical discussion of the principles of mechanics, in the question of teaching students, but he grasped the practical relations which those questions had in every-day work. Gentlemen, I could make a long speech about it, but I would simply repeat what Professor Thurston said, and on behalf of the Society, Professor Thurston, I tender to Mr. Suplee and to his sister, Miss Suplee, our most earnest, heartfelt thanks for this great addition to the portraits we have around us.

Professional papers were taken up at the close of the presentation, the one on "Steel Castings," by Mr. H. L. Gantt, receiving discussion from Messrs. Durfee, Taylor, Platt, Pearson, Royce, Fritz, Almond, and Baker. The paper by Mr. William A. Pike, on "Steam Piping and Efficiency of Steam Plants," was discussed by Messrs. Walworth, Le Van, Coxe, Pearson, Taylor, Sinclair, Ashworth, Thurston, Durfee, Dinkel, and Baker. These discussions proved so interesting that the session closed by Mr. C. H. Manning's paper on the "Method of Manufacturing large Steam Pipes."

Sixth Session. Friday, December 8th.

The Secretary announced the death of Prof. Johann Bauschinger, Munich, Germany, honorary member of the Society. The papers of the morning were that by Mr. J. B. Stanwood, on the "Strength of Rim-joints in Fly-band Wheels;" by Prof. W. A. Rogers on "Cumulative Errors of a Graduated Scale;" by William H. Francis, on a "Modern Disinfecting Plant," which

was discussed by Messrs. Le Van, Woolson, and Jacobus; by W. E. Crane, "Crucible Furnace for Burning Petroleum," discussed by Messrs. Durfee, Henning, Cary, Woolson, and Torrey; by F. A. Scheffler, on "Test of a Boiler using Grates with small Percentage of Openings," discussed by Messrs. Cary, Cartwright, Le Van, Carpenter, and Kent.

Professor Carpenter then read his paper entitled "Experimental Determination of the Effect of Water in Steam on the Economy of the Steam-Engine," which was discussed by Messrs. Le Van, Ball, Jacobus, Barnes, and Scheffler.

A paper by Mr. David Guelbaum, of Syracuse, a non-member, was introduced by the Secretary, on "Theory of Direct-acting Steam-Pumps and its Results," discussed by Professors Sweet and Jacobus.

Topical discussions were then taken up until adjournment. They covered the subjects of "Method for Correcting for the Superheating of Steam in Figuring a Boiler Test," "The Difference in Chimney sizes required for Boilers of ordinary Tubular type and the Water-tube forms," "The best Form of Cylinder Lubricators for Engines," "Safe maximum Speeds for Hoisting in Overhead Traveling Cranes," "The Advantage of using a Circulating Device for the Water in a Marine Boiler, and for grouping the Tubes over the Furnace in such Boiler," "The Advantage of a Forced Circulation of Air to prevent Decay of Work under Floors near the Ground," "The Molecular Change in Metals when their Temperature is Raised or Lowered," "The Factor of Safety in Earthwork Dams," "The Best Form of Straightening Machine," "Sand-blasting as a Method of Producing Surface on Cast-iron on Nickel-plating," "Japanning as a Method of Protecting Steel from Corrosion," "Use of Diametrical Pitch for Large Gears," and "Method of Determining the Cost of Manufactures."

The session closed with a discussion on "Non-conducting Coverings," "The Factor of Safety for Steel Boilers," and "Improvements in the Planimeter for measuring Areas of Indicator Diagrams." "The Progress made in the Use of Aluminium for Castings" was made a matter for extra reference in a supplementary session held after the adjournment of the regular meetings.

The afternoons of the days upon which the sessions were held were left without assignment, for the members visiting New York to attend to personal business affairs, or to make such visits as their inclination and interest might dictate. This policy replaced the usual one prevailing elsewhere, of providing official excursions for the party as a whole.

The policy inaugurated in 1892, of having a light luncheon served in the banquet room at the close of the professional sessions of each morning, was maintained this year also with marked success. It added to the pleasure of the members attending the meetings, and was the means of keeping together those who had assembled for the discussion of papers. Many of the members remained at the house for conversation and social intercourse during the afternoons.

The list of places available for members to visit in New York was as follows:

I. Morgan & Quintard Iron Works.
II. North River Iron Works.
III. Pond Machine Tool Works.
IV. Stevens Institute of Technology.
V. Columbia College, School of Arts, Law, Mines.
VI. Columbia College, School of Medicine (College of Physicians and Surgeons).
VII. Pratt Institute.
VIII. Washington Bridge.
IX. Shipyards and Works; S. L. Moore & Sons Co.
X. World Building; Equitable Building.
XI. Metropolitan Museum of Art.
XII. Museum of Natural History.
XIII. East River Gas Works.
XIV. Ball & Wood Engine Co.
XV. Power Houses, Broadway Cable Road.
XVI. Power House, 125th St. Cable Road.
XVII. The Brooklyn Navy Yard.
XVIII. The Gas Engine and Power Co.

DLVII.

PRESIDENT'S ADDRESS, 1893.

THE USE OF SMALL SIZES OF ANTHRACITE COAL FOR GENERATING STEAM.

BY ECKLEY B. COXE, DRIFTON, PA.

(President of the Society, 1892-93.)

Gentlemen:—It seems proper, on an occasion of this kind, when it becomes my pleasant duty to make my first formal address to the Society, that I should, if possible, call your attention to some matters which, while of general interest to the profession, are in the direct line of my ordinary work.

There is, perhaps, no subject of more importance to us all than the generation of steam; the field is, of course, a wide one, far too extensive to be covered by an address of this kind. I might discuss the question of the water used, its impurities, and the difficulties to be encountered when pure water cannot be obtained; I might discuss the construction of boilers, or the erection of steam plants, or the use of steam at high or low pressure, or the methods of conveying steam and the methods of utilizing it, but before a body such as this, composed of men much more familiar than I am with these questions, remarks from me would, perhaps, be out of place.

But when it comes to the question of the fuel used, I enter upon a field to which my professional life has been devoted. I shall ask your attention this evening to some thoughts upon the use of certain cheaper fuels; viz., the small sizes of anthracite, which are now coming into very general use for generating steam.

Some three years ago I was appointed a member of a commission to investigate the waste of anthracite coal, and the question of the utilization of these small sizes was studied with great care by us. The subject was treated generally in a report made to the State of Pennsylvania in May, 1893, and also in a paper which I read before the American Institute of Mining Engineers at their August meeting (1893), upon an "Automatic Stoker."

When I first remember the anthracite coal business, all the sizes

below stove coa. were considered of little or no value. Chestnut coal was used to a greater or less extent about the collieries for making steam, but much of it was thrown away. It gradually became a domestic fuel, and pea coal took its place for generating steam. This size is gradually coming into use for domestic purposes, and buckwheat coal is taking its place for steam. Two smaller sizes, known as rice and barley, are already being used to a large extent for the same purpose. The value or cost of these coals decreases very rapidly with the size. I append herewith a table showing the size and approximate cost at mines, and the sizes of mesh over which or through which they are made.

TABLE I.

Size.	Made Through.	Made Over.	Approximate Price at Mines.
	Inch.	Inch.	$
Chestnut	$1\frac{1}{2}$	$\frac{7}{8}$	2.75
Pea	$\frac{7}{8}$	$\frac{9}{16}$	1.25
Buckwheat	$\frac{9}{16}$	$\frac{3}{8}$	0.75
Rice	$\frac{3}{8}$	$\frac{7}{16}$	0.25
Barley	$\frac{3}{16}$	$\frac{3}{32}$	0.10

NOTE.—The above meshes are all round punched, and are the standards adopted by Coxe Bros. & Co., at the Cross Creek Collieries.

Another important point to be noted is, that the cost of transportation of the small sizes is less than that of the large. From the mines to tide, pea coal is charged thirty cents, and buckwheat, rice, and barley fifty cents less per ton than the domestic sizes.

One great difficulty encountered in the introduction of these smaller sizes was the fact that, as they were of comparatively little value, but little attention was paid to their preparation, and parties who began to use them did not erect furnaces specially constructed for the purpose, but generally burned them in their old plants, sometimes changing the construction of their grate bars a little.

While the size of the domestic coals made by the different companies is practically the same, there has been a great difference in that of the small coals and in the amount of impurities contained in them. Formerly a large amount of culm was employed where it could be obtained cheaply, but its use is diminishing for the following reason: What was formerly known as culm contained the buckwheat, rice, barley, and in some cases the pea coal. Now, however, the pea, buckwheat, rice, and in some cases the barley

are taken out; and the dust that remains, which forms what is now called culm (although it may be still used by mixing with bituminous coal, or in a special furnace in which it is burned as dust after being thoroughly pulverized, or by making briquettes or compressed fuel), is at present of no value alone and in its natural state for generating steam.

As is well known to all those who burn anthracite, sizing is of great importance. The coal burns only on the surface, and therefore, if the size is not uniform, the smaller particles clog the passages between the larger, and thereby prevent a sufficient quantity of air from passing through the furnace. Some of those who have attempted to use buckwheat, rice, and barley have given them up, because of the great want of uniformity in the results obtained. The difficulty may be due to one of two causes, or both. First, to bad sizing, and second, to the presence of a large quantity of impurities. Experiments we have made have proved that the percentage of carbon when small anthracites are burned under proper conditions, and not the size, is what determines the amount of water evaporated per pound of coal; and that the amount of water evaporated per square foot of heating surface decreases with the size of coal.

Cheap steam is the corner-stone of a large number of our most important industries, and nothing is more important in obtaining it than cheap fuel. The enormous quantity of these small anthracites we are now forced to make in preparing the larger sizes promises to furnish to the northeastern and eastern states a clean and economical fuel, which lends itself to mechanical handling far better than any other known combustible. The only thing that prevents every ton that can be produced from being used is the want of knowledge of its peculiarities and of proper methods of determining its value.

In many parts of Europe coal is sold based upon the percentage of ash; that is to say, coal is guaranteed to contain not over a certain percentage of ash, and the price per ton is reduced a certain number of cents per ton for each additional unit of ash, and in some cases the deduction in the price increases in a geometrical and not an arithmetrical progression, as the percentage of ash increases. It seems to me that it is of the greatest importance to the producer and user of the smaller anthracites that they should be sold on this basis, with an additional guaranty on the part of the producer as to the size.

It is not simply a question of the absolute size of these smaller

coals, but also one of the relative size of the different pieces which make up the aggregate; that is to say, each size should be as nearly uniform as possible. If we consider the coal to be made up of three sizes, Nos. 1, 2, and 3, No. 1 being the largest size, a ton of coal composed entirely of No. 2 would be much more valuable than one composed of the three, although the quantity of No. 1 in it might be greater than the quantity of No. 3. The smaller pieces tend to stop up the interstices between the larger lumps and to check the draft. It is one of the problems in burning small coal to force the air through a grate which will not let the coal drop, and through the body of coal which is more or less compact. Of course, it is not practically possible to have the size of the coal absolutely uniform, but the producer should at least guaranty that a certain percentage of the coal sold as rice should be of such size that it would pass through a mesh of a specified diameter, and over a mesh of another specified diameter. In other words, with the present light I have upon the subject, if I were in charge of a large steam plant, burning, say, 100,000 tons of rice coal per year, I would try to make a contract which would require the coal to contain not over a certain percentage of water and not less than a certain percentage of carbon, or, what is practically the same thing for anthracite, not over a certain percentage of ash; and that a certain percentage of the average daily sample of the coal would pass through a mesh of a certain size and over a mesh of another size, and that the coal should not contain more than a certain percentage of dust—dust being what would pass through a mesh of say $\frac{1}{16}$ of an inch in diameter.

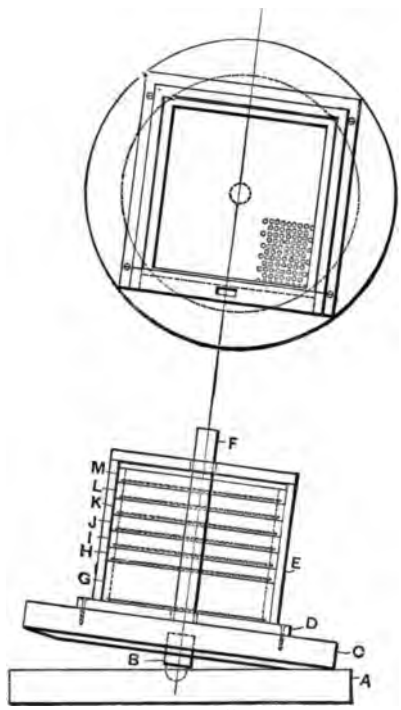
Having reached these conclusions some time ago, I have, with my assistant, Mr. John R. Wagner, been studying the question of how to determine, in a rapid, simple, and effective manner, by methods which do not require scientific work except for the analysis of the coal, the actual commercial value of any shipment of these small sizes.

I shall now proceed to describe our methods for doing this work.

As we are producers to a much larger extent than we are consumers of the small anthracites, although we burn between 100,000 and 150,000 tons per year at our collieries and works for generating steam, we have taken up the subject with the view of controlling our production as well as controlling our consumption, and I shall describe our methods of doing the work, which can easily be adapted by any one receiving coal in cars or boats to their special case.

The foundation of all such work is the obtaining of fair average samples. This is done as follows:

At each of the collieries where the pea, buckwheat, rice, and barley are shipped, samples of each size are taken several times a day from the loading lips as the coal is running into the cars. These are put into four separate bins, one for each size. At the end of the week the coal in each bin is thoroughly mixed, and is then sampled by quartering down until we obtain about thirty pounds, which is sent in a strong canvas bag to the laboratory, thus obtaining from each breaker an average sample of each of the four sizes each week.



SIZING SCREEN.
FIG. 1.

The work of taking samples is under the direction of the coal inspector (one being stationed at each colliery), who inspects all the coal shipped. When cars of small capacity are loaded, of which there are not many now, the samples are sometimes taken from the top of the cars instead of taking them from the lips as the coal runs into the cars. With cars of larger capacity the samples are always taken as the coal runs from the lips, getting in each case a very fair average. In taking samples from the top of the smaller cars, eight or ten ordinary buckets of coal are taken from different parts of the car without paying any attention to the character of the coal at the place it is taken. The coal from the buckets is placed on a platform and quartered down until about one bucketful is obtained, which is put in the bin. In the other case, where the samples are taken as the coal runs from the lips, they are also put in a bucket and carried to the proper bin. Each of the samples obtained may be used for any of the five following operations:

First, Determination of size.

Second, Determination of the slate.

Third, Determination of the ash, involving, in some cases, an analysis of the ash.

Fourth, Determination of the water.

Fifth, Rapid commercial determination of the specific gravity of the coal.

Determination of Size.—The object of this operation is to determine the exact size of coal which makes up the sample, as distinguished from its nominal size. Fig. 1 is a sketch of the apparatus which we employ. It consists essentially of a cubical box, the top and one side of which are open, and into which slides a series of drawers. We generally use five or six. These drawers are wooden frames $1\frac{1}{4}$ inches deep, about 12 inches wide, and 14 inches long. The bottom is a sheet of perforated metal. The box rests upon a circular base of wood, *C*, which is coned at the bottom and has in its center a pin, *B*, with a spherical head, which works in a spherical hole in the center of the wood base, *A*. When the pin is placed in the center hole the box can be revolved easily, and in such a way that the axis of the box describes a cone. This shakes the coal and causes it to move from one side of the screen to the other, and as it can be moved very rapidly, soon sizes the coal. When the drawers have been placed in the box, a stick of wood, *F*, is inserted as shown in the sketch, thus preventing the drawers from sliding out. The size of the perforations which we usually employ are:

For the upper screen, $\frac{7}{8}$ inch round.
For the next lower, $\frac{9}{16}$ " "
For the next lower, $\frac{3}{8}$ " "
For the next lower, $\frac{3}{16}$ " "
For the next lower, $\frac{3}{32}$ or $\frac{1}{16}$ inch round.

This makes, with the dust, six sizes. Of course, the number of drawers and the size of the perforations can be varied to suit each special case. If the coal is of the ordinary character, we can screen about 15 pounds, or about one-half the regular sample brought to laboratory, which is quartered down to obtain the amount necessary to make the determination. It is not necessary to weigh the part to be tested; it is, however, necessary that the entire sample obtained by quartering down should be screened, otherwise you would not have an average sample.

The coal is poured upon the screen, which is then gyrated. When the operator is satisfied that the coal is thoroughly screened, each drawer is taken out and the quantity in each drawer is weighed, and

by simply adding together these weights, the total weight of the sample is obtained. In this way the average percentage of each size is easily found.

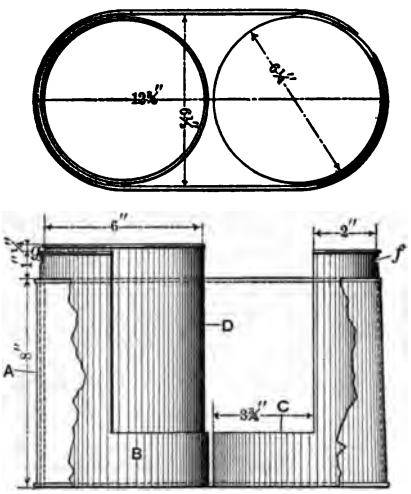
Determination of Slate.—While in large coal the percentage of slate can easily be determined by taking the average sample, weighing out say 100 pounds of it, picking it over by hand and weighing the amount of slate and bone found, it becomes very difficult to do this with the smaller sizes of coal, as a magnifying glass must be used. This is very trying on the eyes and requires a great deal of time. We sometimes resort to it, but only for the purpose of checking our other process. The method adopted depends upon the difference in the specific gravity of the coal and slate. The specific gravity of a piece of coal increases with the amount of ash that the coal proper contains, and also with the amount of bone or slate attached to it.

We first determine by analysis the composition of the coal, bone, and slate from the mine we wish to study, and also find in the usual way the specific gravity of the specimens analyzed. We thus fix a limiting specific gravity, the particles lighter than which may be considered coal, and those heavier, slate or bone; and we make a solution of chloride of zinc of the same specific gravity. This limiting specific gravity, of course, varies with the nature of the coal and the nature of the impurities. The coal from the Lehigh region and some parts of the Schuylkill region is much heavier than that of the other parts of the anthracite field, and the specific gravities of the bone and slate also vary with the locality and vein. We use for our mines a solution with a specific gravity varying from 1.70 to 1.72.

The simplest form of the operation would be to take a large beaker glass nearly filled with the solution of chloride of zinc, and pour slowly into it the coal to be examined, stirring it from time to time. The coal floats and the slate goes to the bottom. When the floating coal, which is generally much greater in quantity than the slate, begins to cover the top of the vessel, a portion of it is skimmed off and put in a dish; and this is continued until all the coal to be tested has been poured into the vessel. The remaining coal which is floating is skimmed off with a great deal of care, the chloride of zinc solution poured off, and the slate placed in another vessel. The coal and slate are then washed, dried, and weighed, and the percentage of each calculated. This, of course, does very well if only a few determinations are to be made, but where it is necessary, as

with us, to make a large number of tests, an apparatus, devised to simplify the work, may be used with advantage. Fig. 2 is a sketch of it.

It consists of a tin vessel, *A* (which the solution of this density does not attack to any great extent), shaped like an ordinary wash-boiler; or, more properly, two one-half cylinders joined by a rectangular parallelopipedon. The depth of the vessel is 8 inches, the extreme length is 12¾ inches inside, and the width 6½ inches where the sides are parallel. The two cylinders, *B* and *C*, the bottoms of which are closed by a wire gauze of No. 24 wire mesh, for 2 inches above the wire gauze are complete, but, higher up, a little more than one-half of the circumference is cut away. These are placed in the outside case, *A*, with the open sides facing each other. Inside of the left-hand one, *B*, is placed another tin cylinder, *D*, of slightly less diameter and without any bottom, which when in position extends from the wire gauze up to the top of the main case.



APPARATUS FOR SEPARATING COAL FROM SLATE BY CHLORIDE OF ZINC SOLUTION.

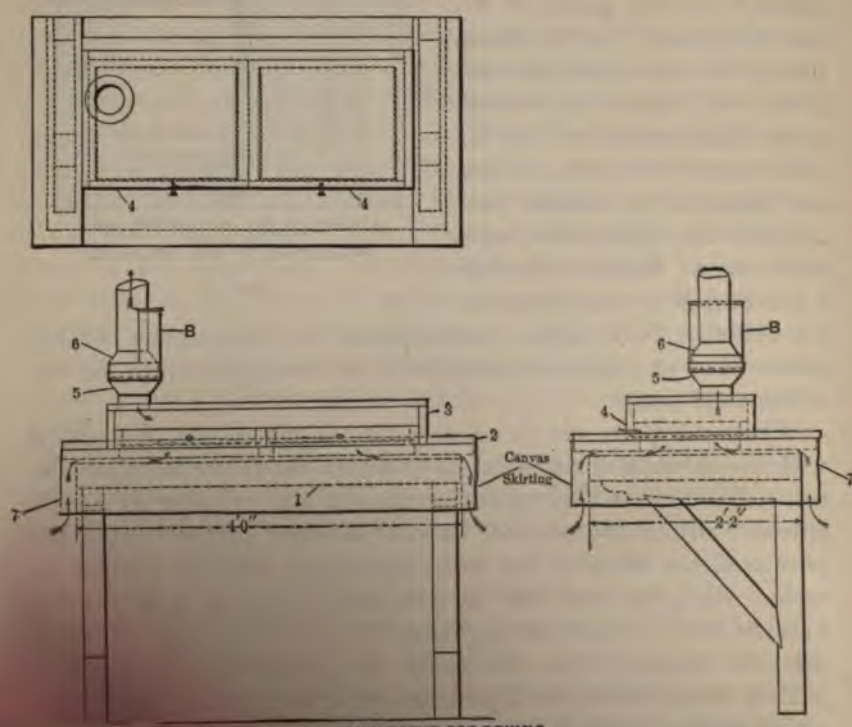
FIG. 2.

When the parts are in these positions, the vessel *A* is filled with the chloride of zinc solution to within about one inch from the top. The coal is then poured gradually into cylinder *D*, and is stirred continuously until all the coal has been poured in, and the pure coal has risen to the top. By raising cylinder *D* about 2½ inches the pure coal can be transferred in it as it is floating to cylinder *C*, when the cylinder *D* is lowered into *C*, while the slate remains upon the gauze of cylinder *B*. Cylinder *B* is then raised above the liquid and hung on a nail by the hole *g*, which allows the liquid to drain back into vessel *A*. Cylinder *C* is then raised, with cylinder *D* inside of it, out of the liquid and allowed to drain, the coal resting then upon the gauze bottom; the two are hung together on a nail immediately over vessel *A*.

After two or three minutes draining, the coal and slate are

washed in the cylinders in which they are; that is, the slate in cylinder *B* and the coal in cylinder *C;* cylinder *D* having been removed, unless there is a large amount of coal, in which case they remain together. They are then removed to the drying apparatus, to be described further on, and after drying the coal and slate are weighed, and the percentage of slate calculated.

A number of experiments have shown us that this method of determining the percentage of slate gives results practically as accurate as those obtained by the most careful selection by eye; although it is possible that in the selection by eye some few pieces might change places. It takes but two or three minutes to make a determination, outside of the time necessary to dry the coal, which is from five to six minutes. So that with the single apparatus five or



APPARATUS FOR DRYING WASHED SAMPLES OF COAL.

FIG. 3.

— an easily be made in an hour by any one of fulness.

As we have constantly occasion to dry coal, which in the ordinary way takes considerable time, a simple apparatus was erected to do the work. It consists essentially of a rectangular iron case (see Fig. 3) about 4 feet long, 26 inches wide, and 4 inches high, cast hollow, and as thin as possible. This is kept filled with steam; on top of this is placed a box made of wood, with side openings, on the projecting bottom of which there are two drawers 18 inches long and 16 inches wide. The sides of these drawers are of wood and the bottom of 24-mesh wire gauze. The bottom of the box, which is about one inch thick, covers the whole iron plate, and extends about one and a half inches beyond the edge all around. From the outer edge of the cover a canvas curtain about six inches long is hung. The projecting bottom board is about one inch above the iron plate, leaving space for the hot air to pass through. The bottom of the case is cut out where the bottom of the drawer is covered with the gauze, and the whole box rests upon the plate, so that air passes around and over the plate, and up through the gauze. The top of the box is closed, with the exception of one hole, in which is inserted a sheet-iron pipe, which connects with a suction fan. The result is that a constant supply of air passes around the iron plate, through the holes in the wood cover, and then through the gauze to the suction fan; by means of which a very rapid drying at a comparatively low temperature is brought about. In order to dry the coal and slate from the chloride of zinc apparatus, the pipe between the drying apparatus and the fan is enlarged, as shown in the sketch; so that, by removing the cap on the upper part of the enlarged portion of the pipe, the cylinders *B* and *C*, or *C* with *D* in it, can be inserted in the enlarged portion, the cover put on, and the drying of this coal and slate carried on without interfering with the drying in the other part of the apparatus. This apparatus is very simple and inexpensive, and is very satisfactory and thorough in its work.

While not of very great importance, yet it may, perhaps, be well to give our method of preparing the solution of chloride of zinc, with some details as to its cost.

We use a three-gallon crock, such as is sold in the ordinary china stores, in which we place about five or ten pounds of scrap zinc, which is generally the old zincs taken from our telegraph batteries; but, where this is not on hand, any scrap which can be purchased for about three and a half cents per pound will do. We fill the crock with common hydrochloric acid, 1.20 specific gravity, and cover it

with any kind of lid of crockery. The important point, of course, is to have an excess of zinc, which is not wasted, but which is used in the next operation. When all action ceases, the solution is decanted, and is of a specific gravity of from 1.5 to 1.62, which we increase by evaporation in a common shallow tin dish over a low-heat burner or any other source of heat, to a specific gravity of about 1.75. The solution costs, when thus made, about seven cents a pound.

The zinc solution at the end of the week (when we have made about 40 tests) becomes dark from the coal dust, and we filter it by filling a quart funnel a quarter full of ordinary mineral wool, and allowing the solution to drip through. This is always done on Saturday afternoon, so as to begin the week with a clear solution. To keep up the specific gravity a small quantity of the heavier solution is occasionally added.

Preparation of Sample.—The sample which comes from the colliery in the canvas bag, on reaching the sample room of the laboratory, is divided into two parts, one of which is dried (should it be wet) in the apparatus already described, and then goes to the pulverizer.

The pulverizer consists of a cast-iron cylindrical shell (Fig. 4), 12 inches long and 16 inches in diameter (both inside), through the center of which passes an iron shaft, and inside of it are 16 chilled cast-iron balls, *B*, 3 inches in diameter. The cylinder makes from 70 to 150 revolutions per minute, according to the number of samples to be prepared. Along the surface of the cylinder, and parallel to the axis, runs a slot which is $\frac{7}{8}$ of an inch wide on the inside, and $1\frac{1}{2}$ inches wide on the outside. This is closed by a wrought-iron wedge, *D*, which is held by two clamps, *C*, placed near each end. The coal is introduced by bringing the slot to the top and pouring it in, and is removed by bringing the slot to the bottom and allowing it to run out, the cylinder being oscillated backwards and forwards in order to remove all the stuff, which is received in a Russia-iron pan 24 inches long and 18 inches wide. In this way the whole of the pulverized product can be rapidly removed. We generally put about 15 pounds into the pulverizer at once. Running at a moderate speed it takes about $1\frac{1}{4}$ hours to pulverize 15 pounds of pea coal thoroughly; but, by crushing it in a mortar so that the coal will pass through a $\frac{9}{16}$ round mesh, this time can be very materially reduced. Buckwheat, rice, and barley are pulverized without any previous hand crushing. Rice and barley require only about three-quarters of an hour.

The pulverizer is a very simple, cheap, and effective machine. The surfaces of the slot are planed, as is also the wedge, making a perfectly tight joint, and not allowing any dust to escape.

The pulverized coal received in the pan is quartered down to about one pint, which is screened on a No. 60-mesh sieve, 90% generally passing through; and that which does not pass through is crushed finer by hand on a rubbing plate, requiring but a few minutes.

The anthracite coal of the Lehigh region, containing on an average not more than 5% of volatile combustible matter, might be pul-

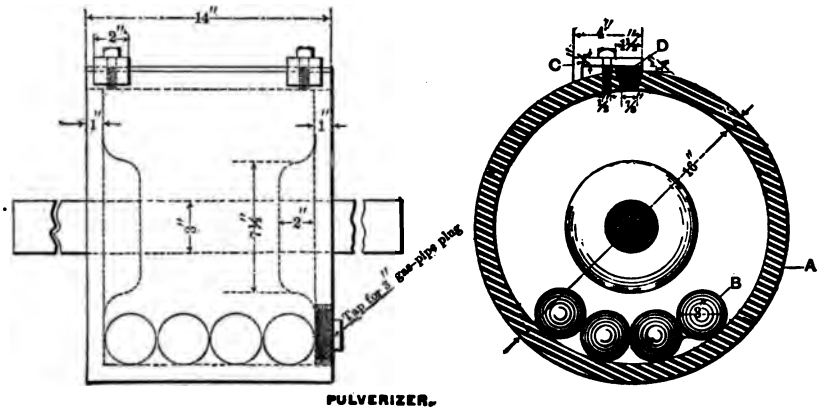


FIG. 4.

verized so as to pass through a No. 80-mesh sieve, which would shorten the time of ash and sulphur determinations, especially if, for the incineration in the former case, oxygen be not used; but as we use oxygen, and only occasionally make sulphur determinations, we only make the sample fine enough to pass through a No. 60-mesh sieve.

Where no machine is available for pulverizing, and the work must be done by hand, it is important to crush the whole sample until it will pass through a sieve of No. 4-mesh, quarter down, then crush the retained portion until it will pass through a No. 10-mesh, then quarter and crush the retained portion until it will pass through No. 24-mesh, and so on until the last sample will all pass through a No. 60-mesh. But no effort should be made to shorten the work by crushing the sample roughly, and screening it over a 60-mesh, and using what passes through for analysis. Our attention having been called to the discrepancies in analysis where we

attempted rapid crushing in this way, we made a series of careful experiments (the results of which are given in the following table), to satisfy ourselves as to the facts. The first column gives the sizes of coal, the second column gives the percentage of ash when the proper method of sampling is adopted, the third column gives the results when the rapid method of sampling was adopted, and the fourth column gives the difference in each case.

RESULTS OF COMPARISON OF TWO METHODS OF OBTAINING THE POWDERED SAMPLES FOR ANALYSIS.

Size.	Method "A." Per cent. of ash.	Method "B." Per cent. of ash.	Difference per cent.
Lump coal	5.00	6.65	1.65
Broken coal	7.52	9.97	2.47
Egg coal	8.50	11.12	2.62
Stove coal	8.39	10.15	1.76
Chestnut coal	9.50	12.87	3.37
Pea	11.45	14.95	3.50
Buckwheat	9.87	11.27	1.40
Rice	13.85	17.95	4.10
Barley	13.05	23.95	10.90

It will be observed that the difference was as much as 84% in the last case. The reason of this is probably that a large portion of the impurities in coal, such as iron pyrites, silica, and substances of a clayey nature, occur in very light incrustations upon the surfaces of fracture of the coal. During the crushing a great deal of these very thin films is rubbed off in powder, so that the fine dust which passes through the 60-mesh sieve contains a very much larger portion of these impurities than the average coal. This is especially the case with the iron pyrites, which is often found in films scarcely thicker than tissue paper, which, during the crushing of the coal, is reduced to a very fine powder.

Although we have not yet made a determination of the sulphur in the case where the 84% difference occurs, I should expect to find a much higher percentage of it in the sample prepared in the wrong way than in the sample prepared in the right way.

Proximate Analysis of the Coal.—For commercial purposes, which is the point of view from which I am treating this subject, only a proximate analysis of the coal is generally necessary; that is, an analysis by which the moisture, volatile combustible matter, and ash are determined; the carbon being determined by difference. It is necessary only occasionally to determine the sulphur, and more

rarely the composition of the ash, as the sulphur and the composition of the ash do not vary in the coal from one colliery as much as the percentage of ash. It would not be necessary to determine very often the moisture and volatile combustible matter were it not that, without them, the amount of fixed carbon cannot be arrived at.

The sample, which has been pulverized and quartered down until it will about fill a pint measure, is sent to the analytical room, where it is subjected to the following operations if all be necessary:

First, Determination of ash.

Second, Determination of moisture.

Third, Determination of volatile combustible matter.

Fourth, Determination of sulphur.

Fifth, Analysis of ash.

Ash Determination.—The sample to be subjected to analysis is obtained by spreading the pint of coal upon a sheet of glazed paper, thinning it out, and then dividing it by cross lines into a number

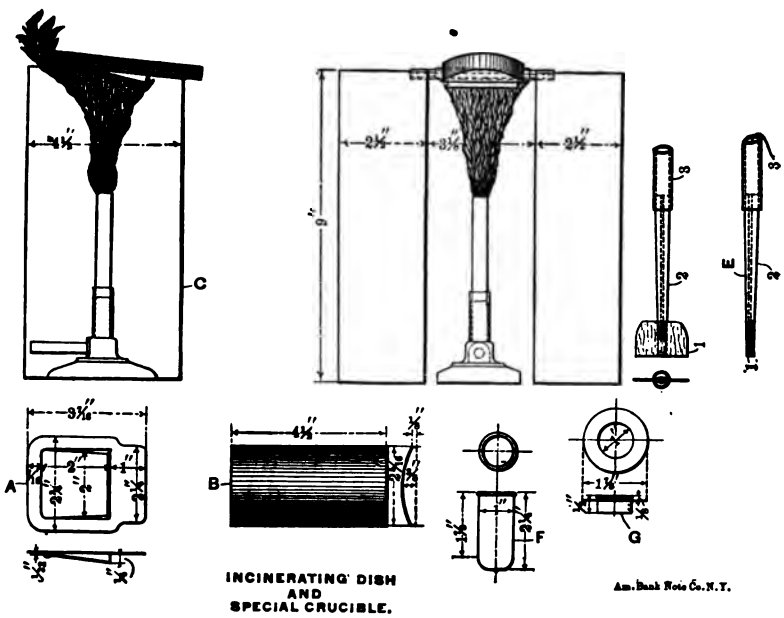


FIG. 5.

of divisions, and from each of these a small quantity is taken, making up the two grams usually used. The incineration is done in light platinum dishes of special shape. (*A*, Fig. 5.) It is practi-

cally a thin sheet of platinum $3\frac{7}{16}$ inches long by $2\frac{3}{4}$ inches wide, with a central depression 2 inches square, and $\frac{1}{32}$ of an inch deep at one end, and $\frac{1}{4}$ of an inch at the other. This is covered with a lid made of thin sheet platinum, $4\frac{1}{2}$ inches long, and bent as shown at *B*. The dish is supported on two platinum wires laid across between two ordinary firebricks, between which a Bunsen burner is placed (*C* and *D*, Fig. 5). The dish can be slightly inclined by allowing the right-hand wire to drop into a crevice about $\frac{1}{4}$ inch deep in the firebrick. The lid and end surfaces of the dish form inclined surfaces which deflect the flame of the burner, and cause it to travel to the left-hand corner, thus creating a slight suction at the end of the lid, and increasing the amount of air passing over the incandescent coal. The under side of the lid, being polished platinum, reflects the heat down upon the coal, and increases its incandescence, thereby hastening the burning out of the carbon. We tried a lid of Russia iron, but, instead of increasing the incandescence, it absorbed the heat and retarded the operation.

One of the great advantages of the shape of the dish is that the sample can be spread in a thin layer, and the current of air or oxygen directed over it, and also that the incandescent mass can be easily stirred, the lid being conveniently lifted with a pair of forceps.

With these dishes, when incinerating two grams and using natural draft, the operation is completed in about one hour, and the coal need only be stirred two or three times. For making ash analysis, we can incinerate five or six grams in the same way in very little more time, as the surface exposed to the action of the air is so great. In an ordinary platinum crucible, with air, it would require from three to four hours for one gram. Of course, I am only talking of anthracite; bituminous coal burns very much more rapidly when treated by the ordinary method. In order to hasten the process, which is a very important point where so many analyses have to be made, we use oxygen, which we obtain in tanks, at a very high pressure, from the New York Oxygen Company. A tank containing 20 cubic feet of gas (at atmospheric pressure) is sufficient for from 75 to 100 analyses.

With oxygen the operation is performed as follows:

After the coal and dish have been heated to redness, a gentle stream of oxygen is allowed to pass over the coal for twenty minutes, after which it is transferred to the other apparatus, immediately alongside, where the same operation is allowed to continue for

twenty minutes more, with a current of air passing under the lid. By having these two apparatuses, one man is able to make about twenty ash determinations of two grams each in ten hours. The flame, as shown in *C*, Fig. 5, is deflected to the left, and on the right-hand side comes to about the lower corner of the dish. The dishes were made to order by John C. Entriken, Malvern, Pa., and weigh from 21 to 22 grams each. The dimensions are all given in Fig. 5. The lids, *B*, are $2\frac{5}{8}$ inches by $4\frac{1}{2}$ inches, No. 40 *B. & S.* gauge, and weighing from 13 to 15 grams each. The wires which support them are also of platinum, 4 inches long and $\frac{1}{16}$ of an inch in diameter, weighing about 4 grams each. The two dishes with which we have already made over 800 determinations are still in fair condition. The right-hand end of the dish is made deeper, first, to avoid loss in consequence of the material sliding down when it first becomes heated; and, second, to give more inclination to the under side of the dish, and thus deflect the flame of the burner to the left. The oxygen is brought from a tank by an ordinary india-rubber tube (Fig. 5, *E* 3), which is terminated by a clay pipe (a piece of tobacco pipe), at the end of which is placed a piece of platinum foil, flattened out as shown in *E*, Fig. 5. This is merely wrapped round and flattened with the fingers, its object being to distribute the oxygen uniformly over the whole width of the dish. We find in our laboratory (where the pressure of gas is low) that it is much easier to incinerate the coal in the flat dish than in a crucible, for the temperature produced by the burner, when the supply of air is restricted, as it must be with low-pressure gas, is hardly great enough to incinerate in a crucible, while there is no difficulty with the dish, and hence results in economy of gas consumption.

Moisture Determination.—This is done in a very simple apparatus, constructed as follows:

Two 1-inch boards, 12 inches wide and about 8 inches high, with the upper corners rounded off, form the back and front. In each of these boards there is cut an opening 10 inches wide and $4\frac{1}{2}$ inches high, also rounded, in which there is placed a copper muffle made of sheet copper $\frac{1}{16}$ of an inch thick, simply bent in the form desired and riveted at the joint, and open at both ends. This is tacked to the two boards. Both the front and back of this muffle is closed by a piece of asbestos board supported on two tacks. Another piece of asbestos board is bent around so as to form the sides and top, leaving a space between it and the muffle about an inch wide.

Underneath the muffle are four very small gas-jets, formed by making four small perforations in $\frac{1}{8}$ inch gas-pipe. These give the heat which keeps the muffle hot. To remove the products of combustion resulting from these jets, four holes, $\frac{3}{8}$ inch diameter, are cut in the top of the asbestos casing. A thermometer passes through the top of the muffle and the asbestos board top, and shows the heat in the muffle. The piece of asbestos board at the back is not taken off; that in front is removable, in order to allow the sample to be put in. By means of these gas-jets the temperature is kept at about 225° Fahr., and remains remarkably constant with but very little attention. Any ordinary mechanic can construct this apparatus in a few hours.

The sample when weighed out is placed in the muffle, and remains at a temperature of 225° Fahr. for about one hour, experience having shown us that this time and this temperature are best suited for determining the moisture in our anthracites. Notwithstanding the extreme simplicity and low cost of the apparatus, it gives results which are very satisfactory.

Determination of Volatile Combustible Matter.—For this determination, the method usually adopted is to heat the sample in a covered platinum crucible for $3\frac{1}{2}$ minutes, by an ordinary Bunsen burner, and then, without cooling, for $3\frac{1}{2}$ minutes more with a blast burner. We find it very convenient to adopt a crucible of special shape, provided with a flanged sleeve (*F* and *G*, Fig. 5). The crucible is 1 inch in diameter, $2\frac{1}{4}$ inches deep, and cylindrical for a length of $1\frac{7}{8}$ inches. It is wire-rimmed at the top, and fits loosely into the sleeve, *G*, which bears against the rim. The sleeve is beaded on top to keep it cylindrical, and $\frac{1}{8}$ of an inch below the top of the sleeve there is a thin annular disk of platinum, which is fused to the other part of the sleeve. The crucible, with this thin sleeve, is supported by a platinum wire triangle placed over a clay cylinder, and which in its turn is supported by a ring of a filter stand at a height suitable to the height of the burner. The clay cylinder is $3\frac{1}{4}$ inches high, with a hole 2 inches in diameter at the top, and tapering down to $1\frac{3}{16}$ inches at 1 inch from the bottom, and from there down widening out again to $1\frac{1}{2}$ inches (Fig. 6, left end of hood). The sleeve prevents the crucible from slipping through, and from tilting over on the wire triangle or the holes in the desiccator. It also helps in heating it to the highest possible temperature for the full length, as it baffles the flame. This sleeve is also very convenient for lifting the crucible from the

cylinder into the desiccator, and from thence into the scale pan, as the operator simply grasps the disk of the sleeve with an ordinary pair of forceps. It weighs 9 grams. We also use the same crucible and sleeve for fusions when making complete analyses of the ash.

Determination of Sulphur.—This is done by Eschka's method, with one exception; that is, we use a different form of crucible, in the shape of a light, small-lipped and wire-rimmed beaker of platinum, holding 100 cc., or about three times as much as the ones usually employed. It is 2¼ inches high, 2 inches in diameter, and tapering slightly toward the bottom, and weighing 33 grams. By using a larger crucible all the operations of oxidation, acidulating, oxidizing with bromine, and driving off the excess of the latter, etc., can be performed without transferring or washing into a beaker, thus saving considerable time.

This method is described in the following periodicals:

Rothe, *Mitheilungen Königlichen Technischen Versuchsanstalten*, 1891, page 107; *Stahl und Eisen*, Vol. XII., page 31.

F. Hundeshhagen, *Journal Analytical and Applied Chemistry*, Vol. VI., page 385; *Chemiker Zeitung*, Vol. XVII., page 454; *Engineering and Mining Journal*, LIV., page 320; *Berg und Huttenmannische Zeitung*, Vol. LII., page 141; *Journal Society of Chemical Industry*, Vol. XII., page 465; *Chemiker Zeitung*, Vol. XVI., page 1070.

James O. Handy, *Engineering Record*, Vol. XXVI., page 373 (few lines.)

Analysis of the Ash.—This, of course, is a comparatively long operation, and is done by the ordinary method.

Rough Determination of Specific Gravity.—This, though a comparatively simple, and to all appearances rough, operation, promises to be of value to producers and users of anthracite coals. The apparatus we employ is constructed as follows:

It consists of four parts: an ordinary sheet-iron bucket such as is used about the works for sand or ashes, and holding about 25 lbs.; an ordinary wash-tub; a Fairbanks' market beam scale; and a cylindrical tin pan about 14 inches in diameter and 7 inches deep. The apparatus is shown on the right-hand side of the view of the laboratory, close to the window (Fig. 6).

The weighing beam is attached to the post by means of a small

crane, which throws it out from the post. The ordinary hook used for suspending material to be weighed carries a yoke from which the tin pan is suspended by two wires. The bucket is also hung on the same hook.

The tub is then filled with water until the tin pan is covered, and the whole is then balanced by means of a weight hung on the outer end of the beam, which weight remains fixed. The poise used for weighing moves along the beam in the ordinary way. The beam is divided into pounds and quarter pounds, each notch representing one quarter pound when the ordinary poise is used. In order to weigh to $\frac{1}{100}$ of a pound, a little rider is made of sheet metal and of such a weight that each notch represents $\frac{1}{100}$ of a pound; so that by placing the poise in the notch corresponding to the even pound which is just less than the actual weight, and then moving the rider until the beam balances, and reading off the number of notches, the actual weight in pounds and $\frac{1}{100}$ pounds is obtained. If, for example, the rider is in the second notch to the right hand of the 10 pound notch, it would read $\frac{42}{100}$ pounds; that is, the 10 pounds would represent 40 notches, which with the other two would make 42. If it was at two notches to the right of 17 pounds, it would represent $\frac{70}{100}$ pounds, or 4 times 17 plus 2.

When the average specific gravity of a shipment of coal is to be determined, specimens are taken from all parts until a fair average sample is obtained. This is spread on a platform, and quartered down until the material is reduced to about 20 pounds, which is then freed from dust, dried in the apparatus shown in Fig. 3, and put into the bucket. It is not necessary to take any specific quantity, simply a bucketful. The bucket is then hung on the hook and the weight of the coal obtained, everything else having already been counterbalanced. When this has been done, the coal is poured slowly from the bucket into the pan in the tub, and after being stirred so that all the air escapes (this is important), the weight in the water is taken; deducting from the weight of the coal in the air the weight of the coal in the water gives the weight of the water displaced; dividing the weight of the coal by the weight of the water displaced gives the specific gravity of the coal.

Of course, the calculations are the same as in ordinary methods of determining the specific gravity, but the operation can be performed rapidly and with a large quantity of coal.

There seems to be no doubt that there is a close relation between the specific gravity of coal and its percentage of ash. Mr. Walter

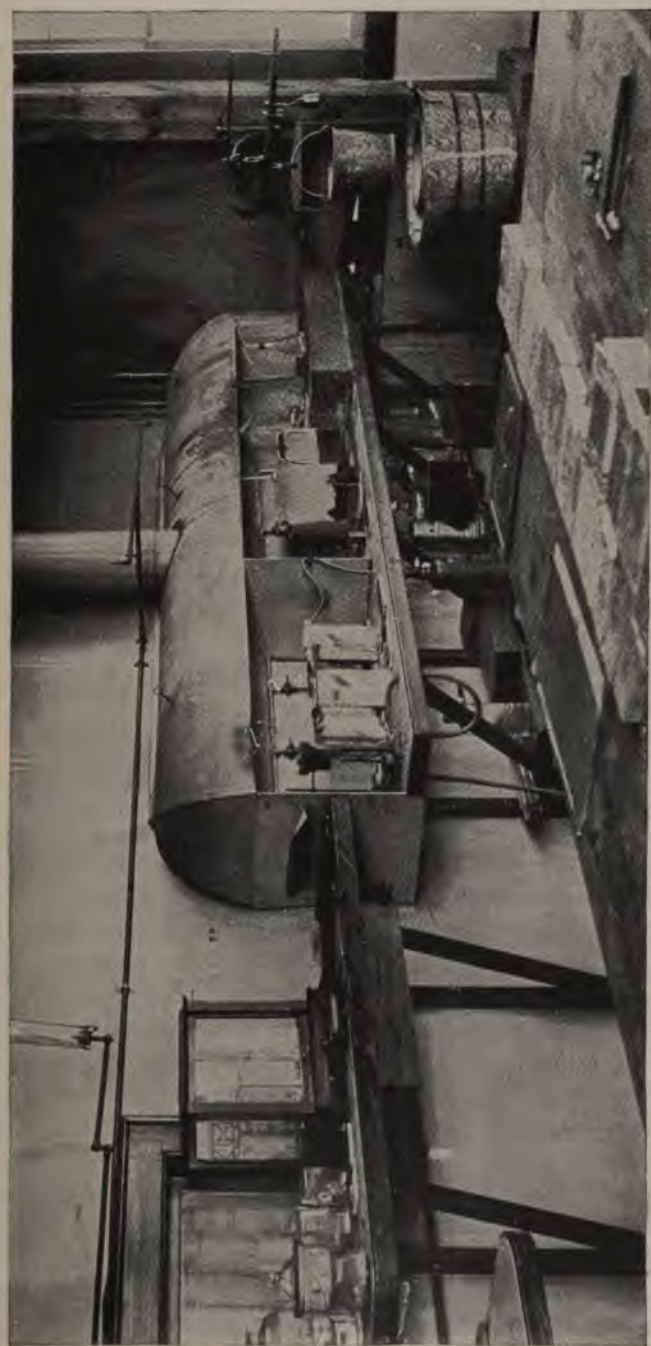


FIG. 6.

R. Johnson, in his celebrated report upon American coals, suggests that there might be such a relation, but gives no figures to establish the fact. A careful study of a great number of analyses of coal and determinations of specific gravity has led us to believe that, although our experiments are not as yet absolutely conclusive, there is a strong probability that, for a given size of coal from the same colliery under ordinary circumstances, the determination of the specific gravity of an average sample will give very nearly the same percentage of ash as will be determined by analysis, although the relation may not be exactly the same for different mines or for different sizes of coal.

If the specific gravity and percentage of ash, in any sample of coal below egg size, is known, the percentage of ash in any other sample of the same size coal, and from the same colliery, can be satisfactorily determined (we are inclined to think) from the specific gravity of that sample, by the following formula:

$$y' = y + (x' - x) \times a;$$

in which

x = the standard specific gravity,
y = the standard percentage of ash,
x' = the specific gravity of coal determined by our apparatus,
y' = the percentage of ash to be determined,
a = a constant for coal from same mine.

It is possible, however, that for the very smallest size from mines where the percentage of iron pyrites is large or very variable, the formula may require some modification. It might also fail if the character of the coal, slate, or other impurities varied very materially in the different veins or in the different parts of a mine, so that the product of the mine could not be considered a uniform one. There is no question, however, that the determination of the specific gravity in this rough way, which can be done by any careful person, is of great value, and would be a very good check on the shipments of coal received, provided it was accompanied from time to time by a slate determination by means of chloride of zinc, and occasionally by an ash determination by analysis.

I do not wish to speak too positively on this subject, because we have not concluded our experiments, and hope later to give more information upon this point.

In the Lehigh region, for any size of coal, we find that, within what may be called the commercial limit of purity, an increase of 0.01 in the specific gravity corresponds to about an increase of $1\frac{1}{2}\%$ of ash; that is to say, that if coal of the specific gravity of 1.54 contained 8% of ash, the same size of coal from the same mine when its specific gravity was 1.56 would contain twice $1\frac{1}{2}\%$ more ash, or 11%.

Fig. 6 is from a photograph of one corner of the analytical room of the laboratory, and is given for the purpose of showing in how little space all the apparatus necessary to carry on the majority of the foregoing operations can be placed. On the extreme right is seen the apparatus for determining the specific gravity. On the left is placed a small but very excellent hood made of galvanized iron (with a ventilating pipe through the roof) in which the operations of determining moisture, sulphur, ash, and volatile combustible matter are carried on.

Almost directly under the pipe is seen the beaker crucible for determining the sulphur, being supported on a sheet-iron frustum of a cone which protects the flame of the Bunsen burner inside. A little to the right of this is seen the drying oven; then comes, under the table, the tank of oxygen, with the hose conveying the oxygen gas to the incinerating dish.

The first pair of firebricks, only one of which is seen, is where the incineration is completed with natural draft. The india-rubber hose passes over one of the bricks which supports the dish, and is protected from the heat by being wound with sheet asbestos. The two bricks in front show the apparatus in which the incineration is done with oxygen, and still further to the left is the apparatus for determining the volatile combustible matter. Of course, while there is nothing specially novel in the arrangement, it is very compact, and shows how little space is necessary for carrying on the work. The balance is placed close to the hood.

We have become so thoroughly impressed with the importance of having the specific gravity of all samples of coal which we analyze, that, whenever the sample sent to the laboratory is large enough, we obtain the specific gravity in the way given above or by means of a flask, and find that it adds very materially to the value of the data obtained by analysis.

It seems to me that in all important boiler tests not only the coal used should be analyzed, and a special determination made of the amount of moisture in the coal when it is actually fired, but

also that the ashes should be carefully analyzed. The sample for both the analyses should be obtained by quartering down all the coal to be used, and quartering down all the ashes made.

I find in the reports of many boiler tests only the name of the coal is given; such as Lehigh egg, Schuylkill stove, Wyoming pea, etc.; in some cases, perhaps, the name of the colliery may be added. Such information is not of very great value, as the quality and purity of these coals vary not only in the same region but in the same mine, from day to day, and any one who wishes to get at the actual facts determined by the tests has not the information necessary to do it. Even if the amount of moisture and volatile combustible matter in the coal is known, the exact amount of carbon burned cannot be determined except by an analysis of the ash. If we know exactly how much coal is placed in the furnace (including what is left on the grate), and how much carbon there is in the ash, we can determine the exact amount of the carbon (originally in the coal) that has been consumed, less the small portion carried up the chimney with the fine ashes.

I have found nearly 60% of carbon in ashes from some boilers; and in some of the boiler tests which have been reported to me not much over 60% of the carbon actually in the coal seems to have been burned.

Under these circumstances it seems to me misleading to say that so much water has been evaporated per pound of combustible. In fact, it is of very little importance how much water can be evaporated under special circumstances with a pound of combustible. What the man who pays for the coal wants to know—and he is the one for whom the profession generally works—is how much water can be evaporated for $1 with different coals; and it is of no advantage to him to know that if he pays twice as much for his fuel he can evaporate 5% more water per pound of fuel, and get a wonderful theoretical evaporation from his boiler. It is the question of how much more you can get for $1 by using a certain coal, and to arrive at this result it is very important to know exactly what kind of fuel is being used, whether it burned up, and whether it was artificially dried before it was put into the furnace in the test.

There is another question of very great importance to users of coal, on which I am not, however, ready to say much, that is, the clinkering. It depends undoubtedly upon two points: the temperature at which the coal burns, and the composition of the ash. The fact that a coal does not clinker may be due to its impurity; that

is to say, the coal may be so impure that it will not make a fire hot enough to melt the ash, although the composition of the ash may be such that were it contained in a pure coal it would melt easily. On the other hand, the coal may clinker because it is very pure and burns at a very high temperature.

It is generally admitted that the more bases there are in the ash the more fusible it is likely to be. It seems probable, from information we have, although it is not very definite, that two coals, each of which burns without giving much trouble on account of the clinkering, may clinker more if they are mixed; the reason being that the composition of their joint ash is more fusible than that of either of the ashes separately.

This subject is one of great importance, and we have taken it up with the view of making further investigations. I would be very much obliged to any one who has analyses of the ashes of any coal, remarkable either for not clinkering or for clinkering badly, if they would send them to me.

In conclusion, I wish to say a few words upon the character of this address.

It does not pretend to treat thoroughly any part of the subject, but simply to state what we are doing, and what we think we have learned. I have no doubt that if I were to deliver this address next year I could add very materially to some of the information that I have given, and develop further some of the ideas that I have ventured to advance; but, on the other hand, I think it almost certain that I would have to revise, and perhaps totally change, some of the views which I have expressed. I am strongly of the opinion that the time has come when it is better for those engaged in professional investigations to give to the public the result of their investigations before they have reached a final conclusion, as it tends to bring out suggestions from others, and to gather facts which will be of great value in the work in hand.

As you have undoubtedly observed, the paper is composed largely of descriptions of what we are doing, or what we have done. No doubt others are doing the same things, in some cases better than we are, and have obtained results of more value than we have, but we give you exactly what we have found. It is not a scientific investigation, properly speaking, but a commercial work, carried on in connection with the business of mining coal, for the purpose of finding out how we can better serve our customers, and make more money in our business.

DLVIII. *

A DEVICE FOR DRILL JIGS.

BY BARTON CRUIKSHANK, BROOKLYN, N. Y.
(Member of the Society.)

THE following simple plan is submitted as leading to good results in the making of drill jigs. This is a well-known device in some shops, but, not being universally used, a brief description of it would perhaps be of interest.

Where two holes are to be placed in the jig in the same flat surface, at a given distance from center to center, they can be quickly and accurately located by the use of hardened steel bushings ground to a known size on the outside, and lapped out to very accurately fit the drills and reamers which are standard for the tits of counterbores in use in the shop. By using the same tit size for several sizes of counterbore the number of bushings necessary for all round work could be considerably kept down. These bushings should be made with a flange or other projection, so that with clamps they may be temporarily fastened to the jig, as shown in accompanying sketch, the distance from the outside of the one to the outside of the other measured with micrometer or vernier calipers, and the holes for the counterbores drilled and reamed.

This is not new nor my own device, but is so handy that it should be more commonly used in shops where sufficient jig making is done to warrant the expense of a few bushings with counterbores to fit.

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* Presented at the New York meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

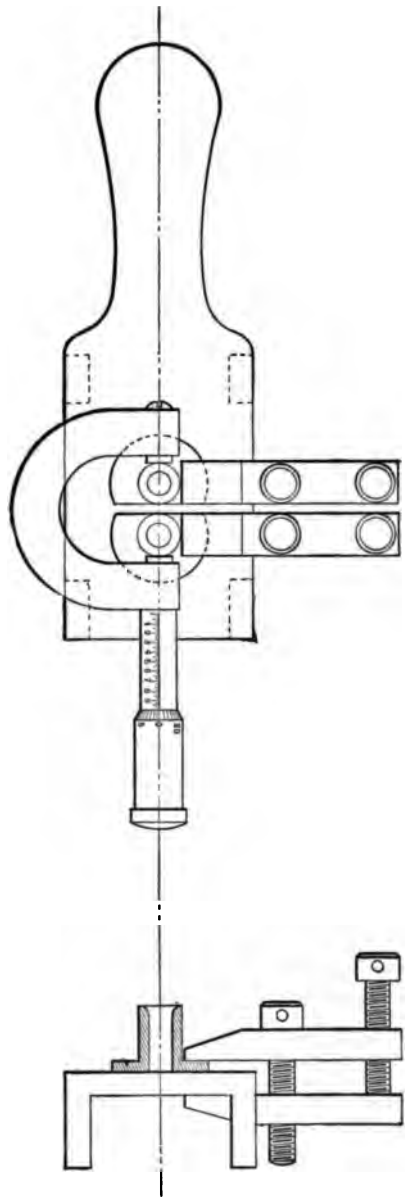

FIG. 7.

DLIX.*

NEW FORM OF PRONY BRAKE.

BY R. C. CARPENTER, ITHACA, NEW YORK.
(Member of the Society.)

THE form of brake described in the following paper was designed by the writer about two years ago, and three brakes, each of about sixty horse-power capacity, have been in use on the experimental engine of the Sibley College laboratories for a considerable portion of that time.

The brake consists, in each case, of a tube of copper, with an elliptical section, which nearly encircles the brake wheel. Outside the copper tube is an adjustable band or belt of sheet steel which prevents the copper tube from expanding. Water pressure is applied to the copper tube by means of hose connections made near the center of the wheel to water pipes; the amount of pressure being regulated by throttling the admission and discharge valves. As the external diameter of the copper tube cannot change, the internal pressure causes the inner surface to rub against the wheel, producing the friction which is used to absorb the work. To prevent the wearing of the inner surface of the copper tube a thin sheet of steel or Russia iron is inserted between the tube and the brake wheel, and connected to the brake, so as not to revolve.

This brake has proved very satisfactory; the work is readily absorbed by varying the opening of the supply and discharge valves, and no trouble has been experienced on long runs from heat or irregular friction of the brake.

In the figures which follow, the principal parts and dimensions are as follows:

Fig. 8.—Elevation of wheel with attached brake. Wheel, 4 feet diameter, 12 inch face; *A*, supply pipe; *B*, discharge pipe; *C*, brake; *D*, sheet-iron liner; *F*, external strap, sheet iron; *G*, adjusting bolts in external strap; *H*, arms of brake riveted to

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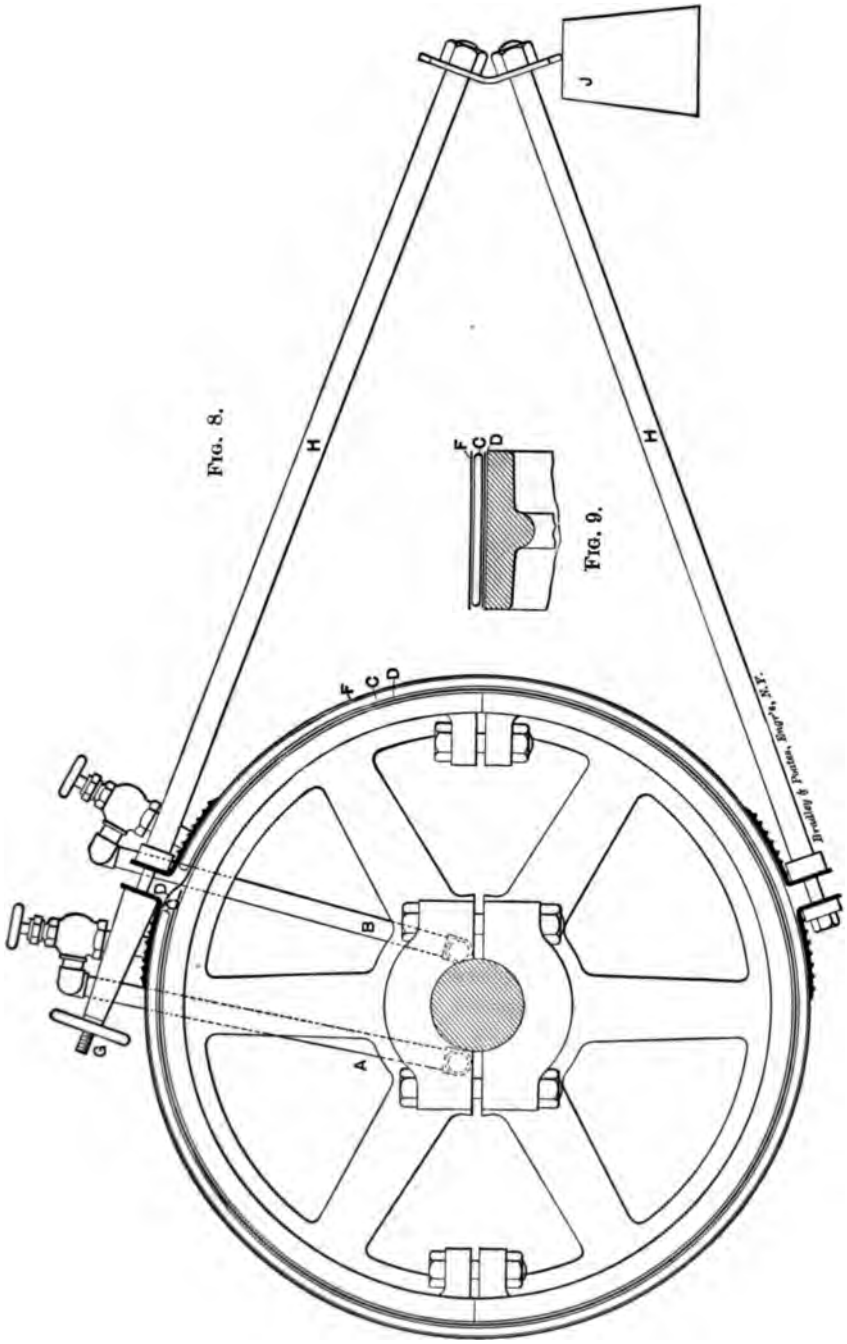

FIG. 8.

FIG. 9.

external strap; *J*, support for brake arms, to be carried on platform of weighing scale.

Fig. 9.—Section of brake and rim of wheel. *C*, brake, elliptical section; larger diameter, 12 inches; shorter diameter, $\frac{3}{4}$ inch; *F*, external strap; *D*, sheet-iron liner.

Another merit, which in some cases may be of little moment, but which was of considerable importance at the time of its erection, was the fact that the first cost was small, the brakes not costing over fifty dollars each.*

AUTOMATIC REGULATING DEVICE.

Prof. C. W. Scribner designed a self-regulating device which I consider of great merit, but which, because of pressure of other work, has not yet been applied to the brakes just described.

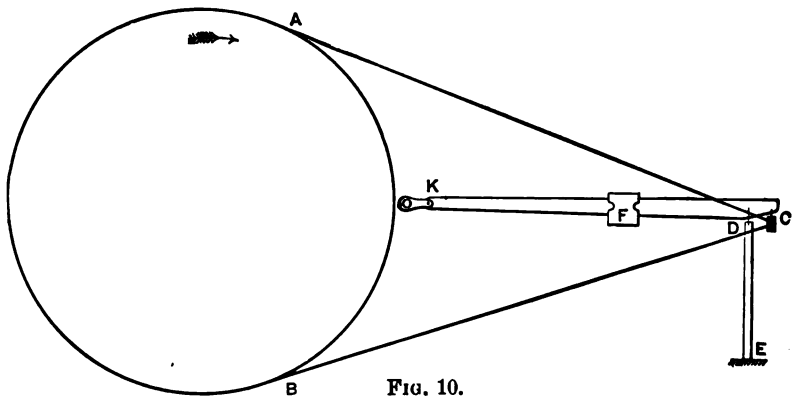

FIG. 10.

There is, however, no doubt of the effectiveness of the design, and its description is therefore here given.

So far as the writer knows, every previous device for automatic regulation of the work absorbed by a brake requires the rotation of the wheel to be such as to tend to move the free end of the brake arm in an upward direction. This is oftentimes objectionable, and sometimes an element of danger. In this case Professor Scribner permits a downward motion of the brake arm, and yet succeeds in securing motion sufficient to adjust the tension on the brake.

* The copper tubes were built by Henry W. Avery, 310 Front Street, New York.

The general method of securing the adjustment is shown in the diagrammatic form, Fig. 10, in which *A C* and *B C* are the brake arms; these rest on the end of a horizontal lever, *K C*. This lever is supported at *D* by a knife edge on the fixed support, *D E*. A movable weight, *F*, is placed on the arm of the lever at any point

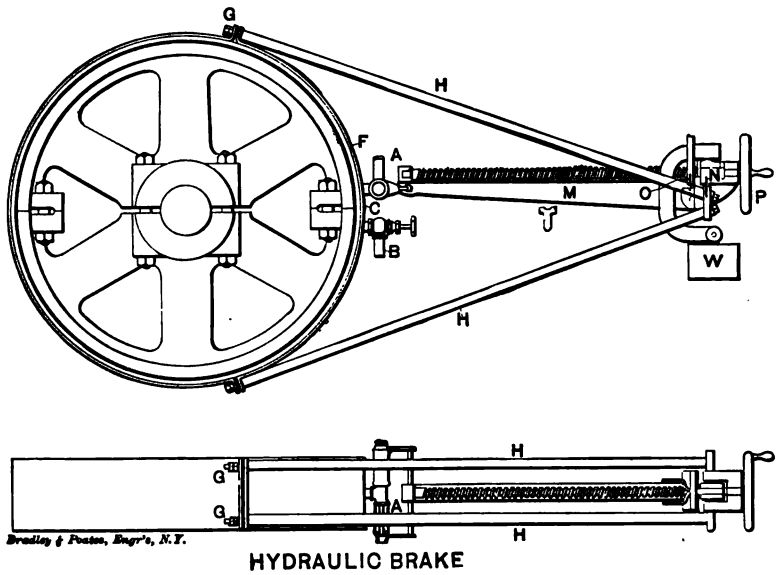

FIG. 11.

needed to give the required brake load. If the brake load is too great, the pressure at *C* is greater than at *F*, and the point *K* of the lever rises; on the other hand, if the brake load is too small, the pressure at *F* is greater than at *C*, and the end *K* of the brake arm falls.

The motion of this point can readily be utilized to decrease or increase the tension on the brake as required. In this case this motion will be utilized to regulate the opening of the supply valves and to vary the pressure on the brake. The weight *F* can also be utilized to weigh the load. The detailed drawings for this purpose are shown in Fig. 11.

The principal parts are as follows:

A, supply pipe; *B*, discharge pipe; *C*, brake; *F*, external sheet-steel strap; *G*, adjusting bolts in external strap; *H*, arms of brake fastened to external strap; *M*, weighing lever; *W*,

weight; *O*, knife edge between lever and stationary support; *N*, knife edge between brake arm and lever; *P*, hand-wheel to adjust load on brake.

It should, perhaps, be mentioned that in the original design for automatic regulator it was expected to support the end of the brake lever on a weighing scale, and utilize the weight *F*, Fig. 10, only for regulating the load. It is now believed that the weight can be used to weigh the load.

DLX.*

A CURVE DELINEATOR.

BY FREDERICK A. SCHEFFLER, NEW YORK CITY.
(Member of the Society.)

EVERY engineer, draughtsman, or mechanic is aware of the inconvenience of drawing arcs or circles of large radii, especially when the location of the center is confined to the space of the drawing-board, or even to the room occupied by the draughtsman. An instrument which is intended to overcome this difficulty has been devised by Theodore Scheffler, M.E., and is shown herewith.

This instrument has been found to overcome the many difficulties involved in attempting to describe arcs or circles from 50 feet radius to 1,000 feet or more. In a few seconds the adjustment of the instrument can be set at such an angle that it would describe an arc of a circle anywhere from 3 feet radius to infinity, and a straight line may be drawn with it. The design of the instrument consists of simply two straight edges, joined at a common center, with a simple device for changing the angle of the straight edges in relation to each other; the device also carrying an attachment for adjusting and fixing a pen or pencil to bear upon the drawing-paper at the point of juncture of the two straight edges. The use of the instrument is based upon the fact that the versed sine of any particular chord is always constant if the radius is constant. To use the instrument, two pins are required to be placed at the extremities of the chord selected, for the points through which the arc is to pass. A Table is given with this paper, giving the requisite angles to set the instrument at, with the fixed chord of 24 inches, and for any radii between 2 feet and 300 feet.

The instrument is operated as follows: By means of the worm-wheel segment *a* (see Fig. 12), the worm *b*, and the graduated wheel *c*, the straight edges may be placed accurately at the de-

* Presented at the New York meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

sired angle v; one revolution of the wheel c changes that angle one degree. The connecting bar d is also graduated by marking off every revolution of the wheel c; when the latter indicates 0 and the edge n of washer plate e is coincident with 0 on the bar d, then the angle v is equal to 0, and the straight edges form a straight line. The penholder h k l should have the position as shown, and l somewhat elevated, so as to give the pen P the proper position. The wheel c being divided into sixty parts, each

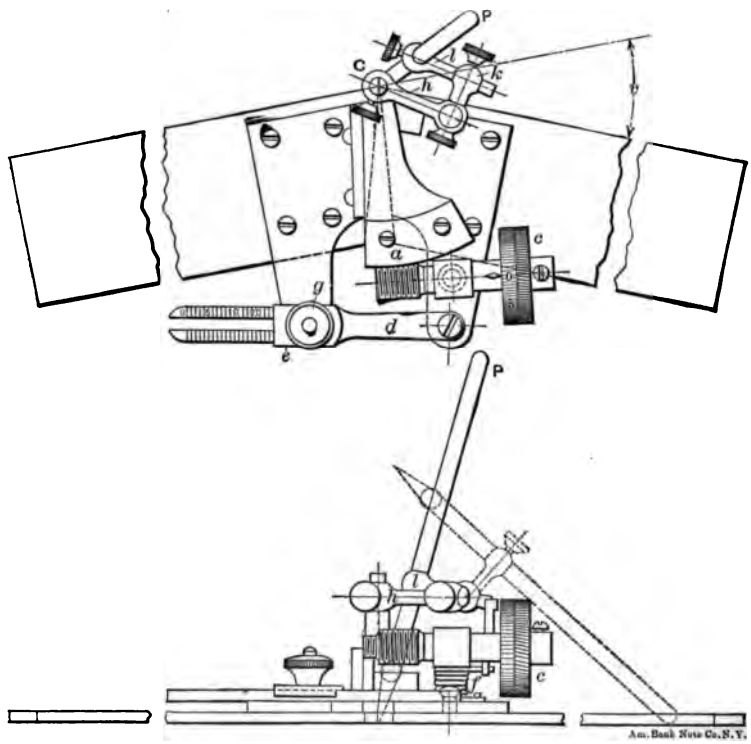

FIG. 12.

part therefore represents one minute. A spring f keeps the worm b tight to a; by disengaging a and b the straight edges may be shifted without turning c. To set the instrument for any particular angle found in the accompanying Table, it is only necessary to turn the graduated wheel c until the requisite number of degrees on the slide and minutes and seconds on the wheel are recorded. To use the instrument for any angle not given in the Table, proceed as follows:

The angle v equals one-half of the angle subtended by the chord; the chord, by the radius = 1, is added to the Table for the purpose of finding the length of chord for a radius not given in the Table. For a curve of a radius 30 feet 6 inches (see Table) set the instrument to the angle 1° 52.7′, set off 24 inches for the chord, place the pins and draw the curve; but if the radius is 30 feet 8 inches, take the next smallest radius, 30 feet 6 inches, and use also the angle 1° 52.7′; take the chord for radius = 1, which is .06555, multiply this by the difference, 30 feet 8 inches − 30 feet 6 inches = 2 inches, therefore .06555

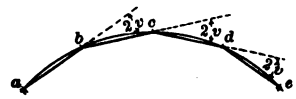

FIG. 13.

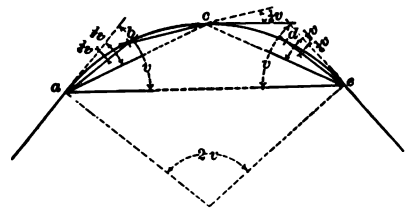

FIG. 14.

× 2 = .1311, or about $\frac{1}{8}$ inch; add this to 24 inches, and $24\frac{1}{8}$ inches is the required chord.

For a curve longer than 24-inch chord, given a radius set to angle 2 v (see Fig. 13), set off points a b c d e, etc., and draw curves with angle v. For a given chord longer than 24 inches, and also for the central angle given, then the following (Fig. 14) will explain: The central angle, or angle of deflection, equals 2 v; instrument at angle v, place it at a and e, produce the tangents, change e to angle $\frac{1}{2}$ of v and produce c, change angle to $\frac{1}{4}$ v and produce from a and c the point b, and from e and c the point d. Draw the curve from a, b, c, d, e. It will thus be seen that with two straight edges, each only 25 inches long, a curve from 24 inches in length, or less, to even 10 feet or more in length can be easily and accurately drawn with *any* radius. The time required to read the above explanation is much greater than the actual time to make the necessary adjustment of instrument. If a chord is given and also the radius, but not the angle, the latter

can be readily determined from the Table, if the chord is even considerably longer than 24 inches. Suppose the chord were 8 feet and the radius 260 feet and adopted chord in the Table 2 feet, then we have the proportion

$$r : 2 : : 260 : 8, \text{ or } r = \frac{2 \times 260}{8} = 65 \text{ feet},$$

r being the radius corresponding to the 2-foot chord, subtending the same angle as the 8-foot chord. Find in Table 65-foot radius, and the angle is 52.89′; this is the angle of instrument; the central angle would be 1° 45.78′. To find the points in the curve proceed as in Fig. 14. If the radius were to be 270 feet, then the radius belonging to the 2-foot chord would be 67.5 feet; this is not in the Table, but 67 feet and 68 feet are, and the corresponding angles are 51.31′ and 50.56′; the difference is .75, and one-half of this difference, = .375, added to 50.56′, = 50.93′, will be the angle for 270-foot radius or for the 67.5-foot radius.

Fig. 15 shows a number of curves running from 2 feet 4 inches radius to 200 feet, all with a 24-inch chord, giving a fair sample of the work which can be done by the instrument.

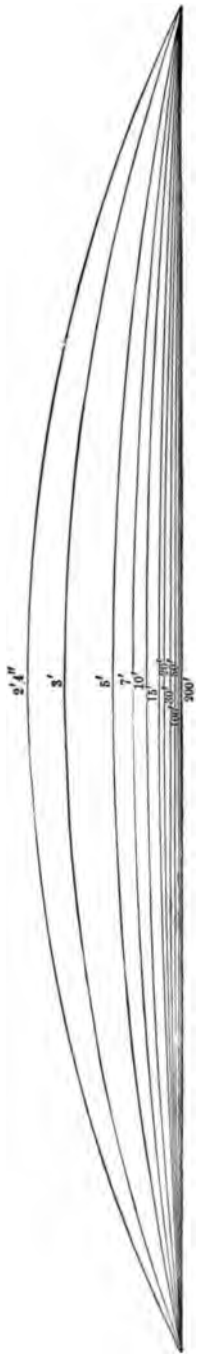

FIG. 15.

For all radii in this table the chords are 24 inches.

Radius.		Angle.		Chord. Radius = 1.	Radius.		Angle.		Chord. Radius = 1.
ft.	*in.*	*deg.*	*min.*		*ft.*	*in.*	*deg.*	*min.*	
2	0	30	0	1.00000		7	8	44.21	.30379
	1	28	41.12	.96000		8	8	37.61	.30000
	2	27	29.80	.92307		9	8	31.17	.29629
	3	26	23.27	.88888		10	8	24.92	.29268
	4	25	22.61	.85714		11	8	18.77	.28915
	5	24	26.60	.82758	7	0	8	12.79	.28571
	6	23	34.70	.80000		1	8	66.95	.28235
	7	22	46.17	.77419		2	8	1.25	.27906
	8	22	01.46	.75000		3	7	55.68	.27586
	9	21	19.42	.72727		4	7	50.25	.27727
	10	20	39.98	.70585		5	7	44.93	.26966
	11	20	03.06	.68571		6	7	39.73	.26681
3	0	19	28.27	.66666		7	7	34.63	.26373
	1	18	55.47	.64864		8	7	29.67	.26086
	2	18	24.51	.63157		9	7	24.82	.25816
	3	17	55.21	.61538		10	7	20.06	.25531
	4	17	27.46	.60000		11	7	15.40	.25263
	5	17	01.12	.58536	8	0	7	10.84	.25000
	6	16	36.00	.57137		2	7	2.01	.24489
	7	16	12.27	.55813		4	6	53.52	.24000
	8	15	49.60	.54545		6	6	45.33	.23526
	9	15	27.90	.53333		8	6	37.54	.23067
	10	15	08.16	.52222		10	6	30.01	.22641
	11	14	47.55	.51064	9	0	6	22.73	.22222
4	0	14	28.65	.50000		2	6	15.77	.21818
	1	14	10.54	.48970		4	6	9.31	.21428
	2	13	53.19	.48000		6	6	2.53	.21052
	3	13	36.54	.47059		8	5	56.26	.20689
	4	13	20.53	.46154		10	5	50.20	.20389
	5	13	05.13	.45283	10	0	5	44.35	.20000
	6	12	50.37	.44444		2	5	38.58	.19672
	7	12	36.13	.43636		4	5	33.27	.19358
	8	12	22.41	.42857		6	5	27.90	.19047
	9	12	99.19	.42105		8	5	22.76	.18750
	10	11	56.43	.41379		10	5	17.78	.18461
	11	11	44.11	.40677	11	0	5	12.95	.18184
5	0	11	32.21	.40000		2	5	8.27	.17910
	1	11	20.72	.39344		4	5	3.72	.17647
	2	11	09.60	.38709		6	4	59.31	.17391
	3	10	58.83	.38195		8		55.02	.17142
	4	10	48.41	.37500		10		50.86	.16901
	5	10	38.32	.36923	12	0		46.81	.16666
	6	10	28.54	.36363		2		42.87	.16446
	7	10	19.05	.35820		4		39.45	.16216
	8	10	9.85	.35294		6		35.31	.16000
	9	10	11.92	.34782		8		31.64	.15789
	10	9	52.25	.34285		10		28.15	.15584
	11	9	43.83	.33802	13	0		24.70	.15384
6	0	9	35.64	.33333		2		21.31	.15189
	1	9	27.68	.32876		4		18.70	.15000
	2	9	19.94	.32432		6		14.88	.14814
	3	9	12.41	.32000		8		11.76	.14634
	4	9	5.08	.31578		10		8.73	.14457
	5	8	57.94	.31168	14	0		5.79	.14287
	6	8	51.00	.30769		2		2.86	.14117

Radius.		Angle.		Chord. Radius = 1.	Radius.		Angle.		Chord. Radius = 1.
ft.	*in.*	*deg.*	*min.*		*ft.*	*in.*	*deg.*	*min.*	
	4		0.03	.13953		9	1	59.60	.06956
	6	3	57.87	.13827	29	0		58.56	.06896
	8		54.57	.13636		3		57.55	.06837
	10		51.93	.13483		6		56.55	.06779
15	0		49.35	.13333		9		55.57	.06722
	3		45.58	.13114	30	0		54.61	.06666
	6		41.95	.12903		6		52.70	.06555
	9		38.41	.12698	31	0		50.91	.06451
16	0		35.00	.12500		6		49.15	.06349
	3		31.68	.12307	32	0		47.44	.06250
	6		28.48	.12121		6		45.97	.06153
	9		25.36	.11940	33	0		44.19	.06060
17	0		22.36	.11764		6		42.63	.05970
	3		19.40	.11594	34	0		41.25	.05882
	6		16.54	.11428		6		39.65	.05797
	9		13.78	.11267	35	0		38.19	.05712
18	0		11.08	.11111		6		36.85	.05633
	3		8.47	.10959	36	0		35.50	.05555
	6		5.91	.10810		6		34.19	.05479
	9		3.43	.10666	37	0		32.92	.05405
19	0		1.01	.10526		6		31.68	.05333
	3	2	58.66	.10389	38	0		30.48	.05263
	6		56.37	.10256		6		29.30	.05194
	9		54.13	.10126	39	0		28.15	.05128
20	0		51.97	.10000		6		27.04	.05063
	3		49.83	.09876	40	0		25.96	.05000
	6		47.76	.09756		6		24.88	.04938
	9		45.73	.09638	41	0		23.86	.04878
21	0		43.75	.09523		6		22.85	.04819
	3		41.83	.09411	42	0		21.86	.04761
	6		39.95	.09302		6		20.90	.04705
	9		38.11	.09195	43	0		19.95	.04651
22	0		36.31	.09090		6		19.03	.04597
	3		35.63	.08992	44	0		18.13	.04545
	6		32.84	.08898	45	0		16.40	.04444
	9		31.15	.08791	46	0		14.74	.04347
23	0		29.51	.08695	47	0		13.14	.04255
	3		27.90	.08602	48	0		11.62	.04166
	6		26.33	.08510	49	0		10.17	.04081
	9		24.79	.08421	50	0		8.77	.04000
24	0		23.27	.08333	51	0		7.41	.03921
	3		21.80	.08247	52	0		6.12	.03846
	6		20.35	.08163	53	0		8.47	.03773
	9		18.94	.08080	54	0		3.66	.03703
25	0		17.54	.08000	55	0		2.51	.03636
	3		16.18	.07920	56	0		1.39	.03571
	6		14.85	.07843	57	0		0.31	.03508
	9		13.54	.07767	58	0		59.27	.03448
26	0		12.25	.07692	59	0		58.27	.03389
	3		11.00	.07619	60	0		57.30	.03333
	6		9.27	.07547	61	0		56.36	.03278
	9		8 54	.07476	62	0		55.45	.03225
27	0		7.35	.07407	63	0		54.57	.03174
	3		6.15	.07339	64	0		53.72	.03125
	6		5.03	.07272	65	0		52.89	.03076
	9		3.91	.07207	66	0		52.09	.03030
28	0		2.80	.07142	67	0		51.31	.02985
	3		1.71	.07079	68	0		50.56	.02941
	6		0.65	.07017	69	0		49.82	.02898

Radius.		Angle.	Chord. Radius = 1.	Radius.		Angle.	Chord. Radius = 1.
ft.	*in.*	*deg. min.*		*ft.*	*in.*	*deg. min.*	
70	0	49.11	.02857	154	0	22.32	.01298
71	0	48.42	.02817	156	0	22.04	.01282
72	0	47.75	.02777	158	0	21.76	.01265
73	0	47.10	.02739	160	0	21.48	.01250
74	0	46.46	.02702	162	0	21.22	.01234
76	0	45.23	.02631	164	0	20.96	.01219
78	0	44.07	.02564	166	0	20.70	.01205
80	0	42.97	.02500	168	0	20.46	.01190
82	0	41.93	.02439	170	0	20.22	.01176
84	0	40.93	.02381	172	0	19.98	.01162
86	0	39.97	.02325	174	0	19.75	.01149
88	0	39.07	.02273	176	0	19.53	.01136
90	0	38.20	.02222	178	0	19.31	.01123
92	0	37.37	.02173	180	0	19.10	.01111
94	0	36.57	.02127	184	0	18.68	.01086
96	0	35.80	.02084	188	0	18.28	.01063
98	0	35.07	·02042	192	0	17.90	.01041
100	0	34.38	.02000	196	0	17.54	.01020
102	0	33.70	.01960	200	0	17.19	.01000
104	0	33.05	.01923	204	0	16.85	.00980
106	0	32.43	.01886	208	0	16.52	.00961
108	0	31.83	.01851	212	0	16.22	.00943
110	0	31.25	.01818	216	0	15.92	.00925
112	0	30.70	.01786	220	0	15.63	.00909
114	0	30.15	.01754	224	0	15.35	.00892
116	0	29.66	.01725	228	0	15.07	.00877
118	0	29.13	.01695	232	0	14.83	.00862
120	0	28.65	.01666	236	0	14.57	.00847
122	0	28.18	.01639	240	0	14.32	.00833
124	0	27.72	.01612	244	0	14.09	.00819
126	0	27.28	.01578	248	0	13.87	.00806
128	0	26.86	.01562	252	0	13.65	.00793
130	0	26.44	.01538	256	0	13.43	.00781
132	0	26.04	.01515	260	0	13.22	.00769
134	0	25.65	.01492	264	0	13.02	.00757
136	0	25.28	.01470	268	0	12.75	.00741
138	0	24.91	.01449	272	0	12.65	.00735
140	0	24.55	.01428	276	0	12.46	.00724
142	0	24.21	.01408	280	0	12.28	.00714
144	0	23.87	.01388	284	0	12.11	.00704
146	0	23.54	.01369	288	0	11.94	.00694
148	0	23.23	.01351	292	0	11.77	.00685
150	0	22.92	.01333	296	0	11.62	.00675
152	0	22.61	.01315	300	0	11.46	.00666

oted by o and e, where e denotes the ratio of cut-

riod of expansion up to the commencement of com-
he steam on the other side of the piston. During
he steam is being expanded on the working side
d on the other side of the piston. The limits of
l be e and $1 - e_1$, where e_1 denotes the ratio of com-

od of compression; the limits of which are $1 - e_1$

o assume that there is an air vessel on both the
uction pipes, the pressures in them being p_1 and
tical height between the water levels in these
ed by h_3, as on the diagram. Let ρ denote the
olume of the liquid. The pressure due to a
f height, l, will therefore be ρl.
ential equations of the piston motion for each of
be:

$$p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\Big\}; \quad oLxLe \ldots\ldots (1).$$

$$\frac{n}{n}p - p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\Big\}; \quad eLxL1 - e_1 \ldots\ldots (2).$$

$$p - \frac{e_1 + m}{1 - x + m}p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\Big\}; \quad 1 - e_1LxL1 \ldots\ldots (3).$$

the distance travelled by the piston, and x
distance to the whole stroke S, so that

clearance volume to the volume of the

moving parts.
admission.
other side of the piston.
mp piston to that of the steam
A^1 of the pump piston.

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April, 1889.

In order to treat the subject in a general way, we shall assume that the steam is worked expansively, the power being regulated by the variation of the degree of expansion.

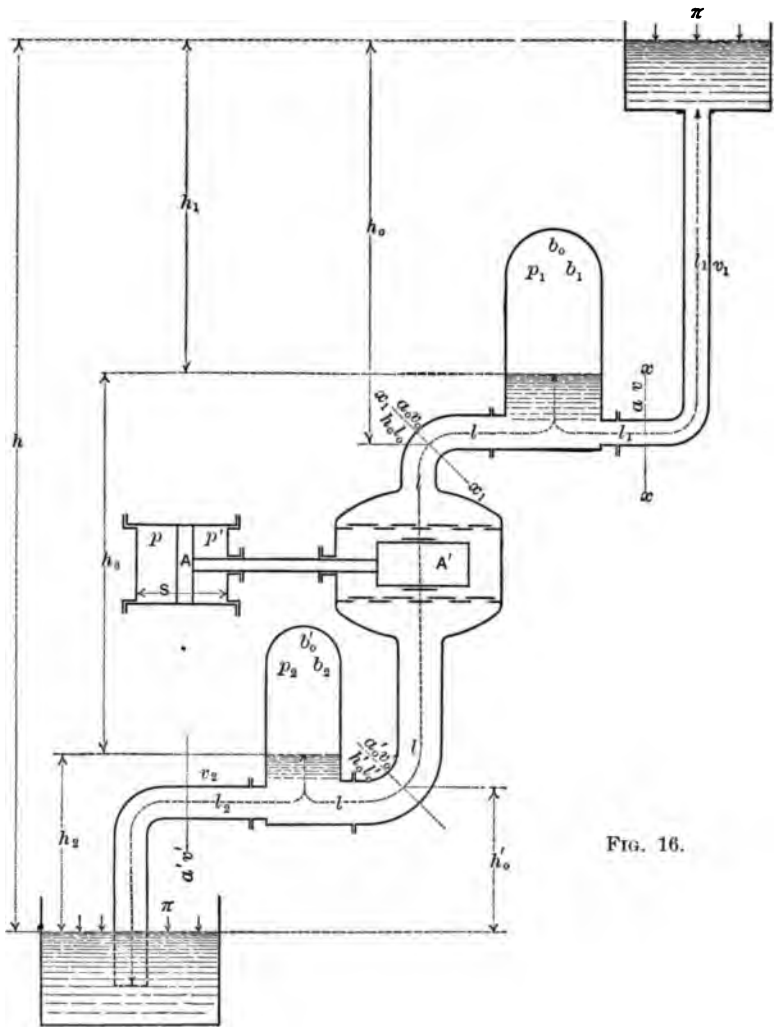

FIG. 16.

I.

The analysis will show whether or not such a system of working is possible.

The motion of the piston may be divided into three periods:

(1) The period of steam admission. The limits of this period

may be denoted by o and e, where e denotes the ratio of cut-off.

(2) The period of expansion up to the commencement of compression of the steam on the other side of the piston. During this period the steam is being expanded on the working side and exhausted on the other side of the piston. The limits of this period will be e and $1 - e_1$, where e_1 denotes the ratio of compression.

(3) The period of compression; the limits of which are $1 - e_1$ and 1.

We shall also assume that there is an air vessel on both the discharge and suction pipes, the pressures in them being p_1 and p_2. Let the vertical height between the water levels in these vessels be denoted by h_3, as on the diagram. Let ρ denote the weight of unit volume of the liquid. The pressure due to a column of liquid of height, l, will therefore be ρl.

Then the differential equations of the piston motion for each of these periods will be:

$$A\begin{cases} \dfrac{d^2s}{dt^2} = \dfrac{gA}{W}\left\{p - p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\right\}; \quad oLxLe \ldots\ldots (1). \\[2ex] \dfrac{d^2s}{dt^2} = \dfrac{gA}{W}\left\{\dfrac{e+m}{x+m}p - p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\right\}; \\ \qquad\qquad eLxL1 - e_1 \ldots\ldots (2). \\[2ex] \dfrac{d^2s}{dt^2} = \dfrac{gA}{W}\left\{\dfrac{e+m}{x+m}p - \dfrac{e_1+m}{1-x+m}p^1 - (p_1 - p_2 + \rho h_3 + R)\eta\right\}; \\ \qquad\qquad 1 - e_1LxL1 \ldots\ldots (3). \end{cases}$$

Where s represents the distance travelled by the piston, and x is the ratio of that distance to the whole stroke S, so that $s = xS$.

m, the ratio of the clearance volume to the volume of the whole piston stroke.

W, the weight of the moving parts.

p, the steam pressure during admission.

p^1, the exhaust pressure on the other side of the piston.

A, the area of the steam piston.

η, ratio of the area of the pump piston to that of the steam piston, so that ηA denotes the area A^1 of the pump piston.

R, the frictional resistance calculated per unit area of the pump piston, so that $R\eta A$ is the total frictional resistance.

Further, denote the velocities of flow of the liquid in different parts of the pipe system by v, v^1, v_0, v^1_0, v_1, v_2, as in the diagram; and the volumes of the air in the suction and delivery vessels by b^1_0 and b_0, when the pump is at rest and the pipes filled with liquid; and the pressure and volume of the air in the vessels at any other time by p_2, b_2 and p_1, b_1.

To deduce equations with reference to the motion of the water, we may notice that the work done by external forces during an interval dt in introducing water into the discharge air vessel is represented by $p_1a_0v_0dt$ and this must be equal to the sum of the following quantities during the same time:

(1) The increase of potential energy of the air within the discharge vessel due to the change of its volume by an amount $-db_1$; i.e. $-p_1db_1$.

(2) The work done in raising a quantity of water having a volume $avdt$ against a pressure $\rho h_1 + \pi$; i.e., $av(\rho h_1 + \pi)dt$.

(3) The increase of kinetic energy of the column of liquid in the discharge main whose mass is

$$\frac{a\rho l_1}{g}; \text{ i.e., } \frac{a\rho l_1}{g} d\left(\frac{v^2}{2}\right)$$

(4) The kinetic energy of the water discharged from the air vessel; i.e., $avdt\rho\frac{v^2}{2g}$.

Hence we have the equation:

$$a_0p_1v_0dt = -p_1db_1 + (\rho h_1 + \pi)avdt + \frac{al_1\rho}{g}\left(\frac{v^2}{2}\right) + av\rho\frac{v^2}{2g}dt \ldots (4).$$

Also expressing the relation between the forces acting on the water in the discharge main—namely, the pressure across the inter face, xx, and the weight of the column of liquid—and the acceleration produced, we have

$$\frac{dv}{dt} = g\frac{p_1 - (\rho h_1 + \pi)}{\rho l_1} \ldots\ldots\ldots\ldots\ldots\ldots (5).$$

Applying the law of Boyle and Mariotte to the air within the discharge vessel, we obtain

$$p_1 b_1 = (\rho h_1 + \pi)\, b_0 \ldots\ldots\ldots\ldots\ldots\ldots (6).$$

The analogous equations for the suction air vessel and main are

$$C \begin{cases} a^1{}_0 p_2 v^1{}_0 dt = p_2 db + (\rho h_2 - \pi)\, a^1 v^1 dt + \dfrac{a^1 l_2 \rho}{g}\, d\left(\dfrac{v^1}{2}\right)^2 + a^1 v^1 \rho \dfrac{v^2}{2g}\, dt & (7). \\ \dfrac{dv^1}{dt} = -\, g\, \dfrac{p_2 - (\pi - \rho h_2)}{\rho l_2} \ldots\ldots\ldots\ldots\ldots\ldots & (8). \\ p_2 b_2 = (\pi - \rho h_2)\, b^1{}_0 \ldots\ldots\ldots\ldots\ldots\ldots & (9). \end{cases}$$

II.

From these equations we ought to be able to deduce all the conditions of motion in steam pumps of the kind here considered. Thus, equations (A) would enable us to determine the variable piston velocity at each point of the stroke; equations (B) would enable us to determine at any point of the stroke the varying pressure, p_1, and volume, b_1, of the air in the discharge vessel and the velocity, v, of the liquid within the discharge main. The analogous quantities for the suction air vessel and main would be obtained from equations (C). Without entering, at present, on the complete solution of these equations, we can deduce from them one general conclusion. For this purpose let us multiply the left-hand side of the equations A by $W \dfrac{ds}{dt} dt$, and the right-hand side by Wds; i.e., $WSdx$. Expressing the fact that the same quantity of liquid must pass across any section during a given interval dt, we have

$$\eta A ds = \eta A S dx = a_0 v_0 dt = a^1{}_0 v^1{}_0 dt \ldots\ldots\ldots (10).$$

In equations (A) replace the quantities $\eta A p_1 dx$ and $\eta A p_2 dx$ by their equivalents from equations (4) and (7). Now integrate between the indicated limits and sum the three equations. The left-hand side of the resulting equation represents the increase of the kinetic energy of the moving masses, W, during a complete stroke, and this increase in the case of direct-acting steam pumps

is always zero at each end of the stroke. The right-hand side will represent the work done during the same time:

(1) By the steam pressure.

(2) By the frictional resistance.

(3) In increasing the potential energy of the air in both air vessels.

(4) In raising the quantity of liquid ηAS.

(5) In increasing the kinetic energy of the whole mass of liquid within the delivery and suction pipes.

(6) In the communication of kinetic energy to the quantity ηAS of liquid raised during suction and delivery.

Also observing that when the motion is settled the quantities (3) and (5) are each zero for a complete stroke, we obtain finally:

$$O = p\left[e + (e + m)\log.\frac{1 + m}{e + m}\right] - p^1\left[1 - e_1 + (e_1 + m)\log.\frac{e_1 + m}{m}\right] - \eta\left[\rho h + \rho\frac{v_1^2 + v_2^2}{2g} + R\right];^{*} \text{ or,}$$

$$p\left[e + (e + m)\log.\frac{1 + m}{e + m}\right] - p\left[1 - e_1 + (e_1 + m)\log.\frac{e_1 + m}{m}\right] = \eta\rho H \ldots\ldots\ldots\ldots\ldots\ldots (11);$$

where h represents the total height through which the liquid is to be raised, and H denotes h + an additional height equivalent to the resistance due to friction and to the kinetic energy of the liquid.

Equation (11) contains nothing specially characterizing direct-acting steam pumps and could have been written *à priori*, without the foregoing discussions, in a similar manner to equations of the same kind referring to ordinary steam-engines and pumps with crank-shafts and fly-wheels. Yet it cannot, as in the case of ordinary steam-engines, be taken as a basis for the determination of the dimensions of a direct-acting steam pump. In direct-acting steam pumps, considered kinematically, the stroke of the piston is not limited, and all the moving masses are subjected only to the free action of external forces and accompanying reactions. In such cases the circumstances are wholly governed by the natural dynamical laws in accordance with the quantity and distribution

* The logarithms used here and in what follows are Naperian logarithms.

of the external forces and masses. Therefore, in the case of direct-acting steam pumps the stroke of the piston and its velocity at every point due to a given steam pressure p, degree of expansion e, and given moving masses, will represent fixed quantities and cannot be chosen arbitrarily. Consequently, applying equation (11) to direct-acting steam pumps, the value of the quantities

$$p^1\left[1 - e_1 + (e_1 + m) \log. \frac{e_1 + m}{m}\right],$$

as dependent on the length of the stroke, and the value of

$$\rho \frac{v_1^2 + v_2^2}{2g} \text{ and } R,$$

as dependent on the piston velocity, must be regarded as fixed but unknown quantities, which are yet to be determined in accordance with the working forces and masses. Hence, if taken separately, equation (11) cannot be applied even to a direct-acting steam pump already constructed, where the ratio η is known, less to one yet to be designed. By reason of this characteristic, only a dynamical equation of motion can be considered as a deciding one for direct-acting steam pumps, and only in connection with such an equation can be made use of equation (11), which refers only to the moment when the piston stops, but not to its motion.

III.

The equations (A) will enable us to obtain the above-mentioned dynamical equation; but, before doing so, we will settle the following questions:

(1) What will be the mean values of the air pressures p_1 and p_2 within the air vessels during one complete piston stroke?

When the suction and discharge air vessels are of sufficiently large dimensions, and especially in the case of duplex steam pumps where there exists a constant flow of liquid from the suction air vessel to the pump, and from the pump to the discharge air vessel, the oscillations of the air pressures p_1 and p_2 within the vessels during each piston stroke will be inconsiderable. The mean value of either of these pressures—namely, that which would

have accompanied a state of equilibrium of all the forces acting on the piston, and a uniform flow of liquid to and from the air vessels (such a state being the middle point of the oscillations) according to the law expressed in the general equation

$$\frac{p}{\rho} + h + \frac{v^2}{2g} = constant \ldots\ldots\ldots\ldots\ldots (12)$$

—will be equivalent to the vertical distance from the liquid level within either air vessel to the liquid level within the corresponding reservoir + the atmospheric pressure height + the height corresponding to the kinetic energy of the liquid moving with its mean velocity. Hence, in equation (A) we can, with sufficient approximation for practical purposes, replace the variable quantity $p_1 - p_2 + \rho h_3$ by its constant mean value

$$\rho h_1 + \pi + \frac{\rho v_1^2}{2g} - \left(\pi - \rho h_2 - \frac{\rho v_2^2}{2g}\right) + \rho h_3; \text{ i.e., } \rho h + \rho \frac{v_1^2 + v_2^2}{2g}$$

where h ($= h_1 + h_2 + h_3$) represents the total height to which the liquid is raised, and v_1, v_2 denote the mean velocities within the discharge and suction mains. If we put $H = h + \frac{v_1^2 + v_2^2}{2g}$ + a height corresponding to the frictional resistance—*i.e.*, a total height equivalent to the indicated work of the steam cylinder as per equation (11)—the variable expression ($p_1 - p_2 + \rho h_3 + R$) in equations (A) may be replaced by its constant mean value ρH.

(2) What kind of masses are to be included in the quantity W introduced in equations (A)?

Let W_1 denote the weight of the solid parts connected with the piston, which moves with a variable velocity $\frac{ds}{dt}$; and let v_0 denote the variable velocity of the liquid column of the length l, and sectional area a_0 enclosed between the air vessels, supposing, for simplicity, that the area is the same, a_0, throughout the length l. Then the first of equations (A) can be written in the form

$$\Sigma m \frac{d^2s}{dt^2} = \frac{W_1}{g} \times \frac{d^2s}{dt} + \frac{1}{g} \rho a_0 l \frac{dv_0}{dt} = (p - p^1 - \eta H)A \ldots (13).$$

But, as before, we have at any moment $a_0 v_0 dt = \eta A ds$; i.e., $a_0 v_0 = \eta A \frac{ds}{d}$: therefore,

$$a_0 \frac{dv_0}{dt} = \eta A \frac{d^2 s}{dt^2} \ldots\ldots\ldots\ldots\ldots\ldots\ldots (14).$$

Introducing this expression for $\frac{dv_0}{dt}$ in the above equation, we obtain

$$\frac{d^2 s}{dt^2} = \frac{gA(p - p^1 - \eta\rho H)}{W_1 + \rho\eta A l} \ldots\ldots\ldots\ldots\ldots (15).$$

This shows that the inertia of the mass of liquid enclosed between the air vessels is equivalent to that of a liquid column of the same length l, and of cross sectional area equal to the area of the pump piston, and moving with the same velocity as the piston. Therefore, we shall always replace

$$W(= W_1 + \rho\eta A l) \text{ by } \rho\eta A L \ldots\ldots\ldots\ldots (16)$$

where L denotes the length of the column of liquid enclosed between the air vessels, together with the length of a column of liquid of cross-sectional area, equal to that of the pump piston and of the same weight as the solid moving parts.

(3) What constitutes the condition of continuity of the liquid column enclosed between the suction and discharge air vessels, that is, the condition that this liquid column forms, together with the piston, one mass which cannot get any motion independently of the piston?

If in a body submitted to the action of external forces, the part on one side of a section has no motion relatively to this section, the resultant of the forces applied to this part will be directed toward this section and not away from it. Hence the resultant of all forces applied to any part of the liquid column between the discharge air vessel and a given section $x_1 x_1$ must be directed towards this section; *i.e.*, opposite to the motion of the liquid column. It follows that each elementary increase of kinetic energy of this part of the column is negative, and, as the velocity is always positive, the acceleration must be negative, for

$$d\left(\frac{m}{2} \times \left[\frac{ds}{dt}\right]^2\right) = m \frac{ds}{dt} \frac{d^2 s}{dt^2}.$$

Hence the condition for continuity of the liquid column is, that the acceleration of any part of the column enclosed between the discharge air vessel and a given section is always negative. The acceleration at the section x_1x_1 in the direction of motion of the liquid is

$$gA\frac{p - p^1 - \eta\rho H}{\rho\eta AL} \times \frac{\eta A}{a_0};$$

also the acceleration in the opposite direction is

$$g\frac{a_0(\rho h_0 + \pi)}{\rho a_0 l_0},$$

where h_0 denotes the vertical distance from the section to the level of liquid in the upper reservoir, and l_0 the length of the part of the column from x_1x_1 to the air vessel. Consequently the condition for continuity of the column of liquid will be

$$gA\frac{p - p^1 - \eta\rho H}{\eta\rho AL} \times \frac{\eta A}{a_0} \leq g\frac{a_0(\rho h_0 + \pi)}{\rho a_0 l_0},$$

or

$$\frac{p - p^1}{\eta\rho H} - 1 \leq \frac{a_0}{\eta A} \times \frac{L}{l_0} \times \frac{\pi \pm \rho h_0}{\rho H} \ldots\ldots\ldots\ldots (17).$$

The + sign on the right-hand side relates to the liquid in the discharge main, and the − sign on the right-hand side relates to the suction main, for which also $h_0 l a_0$ ought to be replaced by the corresponding values $h^1_0\ l^1_0\ a^1_0$. (See sketch.)

Equation (17) provides, as shown later, sure means for obtaining under every circumstance continuity between the pump piston and the liquid column, which, therefore, in the further investigations we shall consider as one whole mass of the weight $\rho\eta AL$ submitted to the action of external forces.

IV.

We shall now proceed to the integration of equations (A). As shown in section III. we may replace $(p_1 - p_2 + \rho h_3 + R)$ by ρH, and W by $\rho\eta AL$. Performing these substitutions, and multiplying the left-hand sides by $2\frac{ds}{dt} \times dt$, the right-hand sides by $2ds$, i.e., $2Sdx$, and integrating, the equations take the form

$$\int_0^{\frac{ds}{dt}} d\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\int_o^x (p - p^1 - \eta\rho H)\, dx; \qquad o<x<e;$$

$$\int_{V_1}^{\frac{ds}{dt}} d\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\int_e^x \left(\frac{e+m}{x+m}\, p - p^1 - \eta\rho H\right) dx; \; e<x<1-e_1;$$

$$\int_{V_3}^{\frac{ds}{dt}} d\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\int_{1-e_1}^x \left(\frac{e+m}{x+m}\, p - \frac{e_1+m}{1-x+m}\, p^1 - \eta\rho H\right) dx;$$
$$1 - e_1<x<1;$$

where V_1 and V_3 are the velocities of the piston when $x = e$ and when $x = 1 - e_1$. Hence

$$\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p - p^1 - \eta\rho H\right] x;$$

$$\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p\,(e+m)\log.\frac{x+m}{e+m} - (p^1 + \eta\rho H)(x-e)\right] + V^2_1;$$

$$\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p\,(e+m)\log.\frac{x+m}{1-e_1+m} + p^1\,(e_1+m)\log.\right.$$
$$\left.\frac{1+m-x}{e_1+m} - \eta\rho H\,(x-1+e_1)\right] + V^2_3.$$

Putting $x = e$ in the first of these equations, we have

$$V^2_1 = \frac{2gS}{\eta\rho L}(p - p^1 - \eta\rho H)\, e;$$

Putting $x = 1 - e_1$ in the second of these equations, we have

$$V^2_3 = \frac{2gS}{\eta\rho L}\left[p\,(e + (e+m)\log.\frac{1-e_1+m}{e+m} - (p^1 + \eta\rho H)\,(1-e_1)\right]; \quad \ldots(18).$$

Hence

$$D\begin{cases}\left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p - p^1 - \eta\rho H\right]x; \ldots\ldots\ldots\ldots o<x<e; \quad (19).\\[2ex] \left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p\left(e + (e+m)\log.\frac{x+m}{e+m}\right) - (p_1 + \eta\rho H)x\right]; \\ \qquad\qquad e<x<1-e_1; \quad (20).\\[2ex] \left(\frac{ds}{dt}\right)^2 = \frac{2gS}{\eta\rho L}\left[p\left(e + (e+m)\log.\frac{x+m}{e+m}\right) - \right.\\ \qquad \left. \eta\rho Hx + p^1\left\{(e_1+m)\log.\frac{1+m-x}{e_1+m} - (1-e_1)\right\}\right]. \\ \qquad\qquad 1-e_1<x<1; \quad (21).\end{cases}$$

The next integration would enable us to determine the time of a complete piston stroke and therefore also the mean velocity of the piston motion. But from equations (D) even in their present state we can obtain the above-mentioned dynamical law of motion for direct-acting steam pumps. In equations (D) let the second factor on the right-hand side be denoted by (T_x) and let $\eta\rho H$ be denoted by (T). By equation (11)

$$T = \eta\rho H = p\left[e + (e+m)\log.\frac{1+m}{e+m}\right] - \\ p^1\left[1 - e_1 + (e_1+m)\log.\frac{e_1+m}{m}\right].$$

Then equations (D) may be written:

$$\frac{\left(\frac{ds}{dt}\right)^2}{2gH} = \frac{S}{L} \times \frac{T_x}{T}; \ldots\ldots\ldots\ldots\ldots (22).$$

The expression $\frac{T_x}{T}$ evidently represents some function $\varphi\,(p, e)$ of the steam pressure and its expansion degree; that is, a function dependent chiefly on the mode of distribution of the steam pressure. Putting

$$H^1 = \frac{1}{2g}\left(\frac{ds}{dt}\right)^2$$

we have the equation

$$\frac{H^1}{H} = \frac{S}{L}\varphi\,(p, e); \ldots\ldots\ldots\ldots\ldots\ldots (23)$$

which gives at every moment the relation between the variable piston velocity and the existing working conditions. If equation (23) exists for the varying velocities $\frac{ds}{dt}$ of the piston, then an equation of the same form certainly exists for the mean velocity $V\left(=\frac{S}{t}\right)$, where S is the complete stroke and t is the time of a complete stroke; but the function of p and e will be different.

Thus equation (23) can be applied to the mean velocity of the piston; and as it contains all the elements of the motion or their equivalents—namely, the path of the piston, its velocity, the mass of all the moving parts, and the external forces—it may be regarded as *the law of motion* for direct-acting steam pumps. It will be more convenient to have equation (23) in the form

$$\frac{S}{L} = 4\,\frac{H^1}{H}\,\Phi\,(p, e)\,; \quad \ldots\ldots\ldots\ldots\ldots\ldots (24).$$

It may be interesting to notice that the dynamical equation (22), like the statical equation (11), could have been written *à priori* without the foregoing deductions. Indeed, it is evident that T_x means the surplus of work developed during the path $s\,(=Sx)$ by the steam over the total resistances relative to one unit of volume of the whole stroke. Therefore, multiplying and dividing the left part of equation (22) by $\eta\rho$, and noticing from equation (11) that $\eta\rho H$ is $= T$, we obtain

$$\frac{\left(\frac{ds}{dt}\right)^2}{2g} - \eta A\rho L = AST_x.$$

Here the left part represents the kinetic energy of the moving parts connected with the pistons at the given point $s\,(=Sx)$, which kinetic energy—as is well known—is equivalent to the surplus of work developed by the steam during the path $s\,(=Sx)$ over the work of all resistances as expressed on the right hand of the above equation. This circumstance may give confidence in the above accepted method of investigation, at the same time suggesting how, in an existing direct-acting steam pump, the values of the function $\varphi\,(p, e)$ or $\Phi\,(p, e)$ can be determined from ordinary indicator diagrams.

V.

The function of equation (24)—$\Phi(p, e)$—can be determined in the following way. Divide the piston stroke into four periods, and let S_1, S_2, S_3, S be the values of s at the ends of these periods. The periods are:

(1) From the beginning of the piston stroke to the period of cut off; $S_1 = Se$.

(2) From the point of cut off to the point where the piston attains its maximum velocity. The value of S_2 is obtained from the condition that when the velocity reaches its maximum $\frac{d^2s}{dt^2} = O$, and therefore from equation (2)

$$\frac{gA}{W}\left[\frac{(e+m)S}{S_2+mS}p - p^1 - \eta\rho H\right] = O.$$

Hence

$$S_2 = S\left[\frac{(e+m)p}{p^1+\eta\rho H} - m\right].$$

(3) From the point of maximum piston velocity until the beginning of the compression of the exhaust steam; $S_3 = S(1 - e_1)$.

(4) From the beginning of compression of the exhaust steam until the end of the piston stroke. Denote the piston velocities at the end of each of these periods by V_1, V_2, V_3, V $(= O)$ respectively. The values V_1 and V_3 have already been found (equation 18), and V_2 can be obtained from equation (20) by putting

$$x = \frac{(e+m)p}{p^1+\eta\rho H} - m;$$

$$V_2 = \frac{2gS}{\eta\rho L}\left[p\left\{e + (e+m)\log.\frac{p}{p^1+\eta\rho H}\right\} - (p^1+\eta\rho H)\left(\frac{(e+m)p}{p^1+\eta\rho H} - m\right)\right]; \ldots (25).$$

Assume for simplicity that the motion of the piston during each of these periods is uniformly accelerated. This assumption is quite correct for the first period, as during it the motive power is constant, and for each of the other periods it is equivalent to replacing the varying accelerations by a constant acceleration—equal to some mean value—which gives the existing velocity at

the beginning and at the end of each of these periods. Therefore, according to the law of uniformly accelerated motion, the times required by the piston to pass through each of these periods will be equal to these periods divided by the arithmetic mean of the velocities at the beginning and end of each period; *i.e.*, will be

$$\frac{S_1}{\frac{1}{2}V_1},\ \frac{S_2 - S_1}{\frac{1}{2}(V_2 + V_1)},\ \frac{S_3 - S_2}{\frac{1}{2}(V_3 + V_2)},\ \frac{S - S_3}{\frac{1}{2}V_3},$$

respectively. Therefore, the mean velocity of the piston will be

$$V = \sqrt{2gH^1} = \frac{S}{t} =$$

$$\frac{S}{\frac{S_1}{\frac{1}{2}V_1} + \frac{S_2 - S_1}{\frac{1}{2}(V_2 + V_1)} + \frac{S_3 - S_2}{\frac{1}{2}(V_3 + V_2)} + \frac{S - S_3}{\frac{1}{2}V_3}}; \ldots\ldots (26).$$

Replacing S_1, S_2, S_3 by their values

$$Se,\ S\left[\frac{(e + m)p}{p^1 + \eta\rho H} - m\right],\ S(1 - e_1),$$

and V_1, V_2, V_3 by their values already found, we obtain from equation (26) the same equation as (24), but with the following form of function $\Phi(p, e)$

$$(27)\quad \left\{\begin{aligned} \Phi(p, e) = {} & \left[p\left(e + (e + m)\log.\frac{1 + m}{e + m}\right) - p^1\left(1 - e_1 + (e_1 + m)\log.\frac{e_1 + m}{m}\right)\right] \left\{\frac{e}{\sqrt{T_1}} + \frac{\frac{(e + m)p}{p^1 + \eta\rho H} - m - e}{\sqrt{T_1} + \sqrt{T_2}} + \frac{1 - e_1 + m - \frac{(e + m)p}{p^1 + \eta\rho H}}{\sqrt{T_2} + \sqrt{T_3}} + \frac{e_1}{\sqrt{T_3}}\right\}; \\ \text{where } T_1 = {} & (p - p^1 - \eta\rho H)e; \\ T_2 = {} & p\left[e + (e + m)\log.\frac{p}{p_1 + \eta\rho H}\right] - (p^1 + \eta\rho H)\left(\frac{(e + m)p}{p^1 + \eta\rho H} - m\right); \\ T_3 = {} & p\left[e + (e + m)\log.\frac{1 - e_1 + m}{e + m}\right] - (p^1 + \eta\rho H)(1 - e_1). \end{aligned}\right.$$

VI.

Thus in direct-acting steam pumps the work done and the circumstances of the motion are expressed by equations (11), (17), and (24). Modifying these equations to make them suitable for general use, we have:

$$\text{(I.)}\ldots\ldots \frac{\eta\rho H}{p} = e + (e + m)\log.\frac{1 + m}{e + m} - \frac{p^1}{p}\left[1 - e_1 + (e_1 + m)\log.\frac{(e_1 + m)}{m}\right];$$

$$\text{(II.)}\ldots \begin{cases} \dfrac{p - p^1}{\eta\rho H} - 1 \leq \dfrac{a_0}{A_1} \times \dfrac{L}{l_0} \times \dfrac{\pi + \rho h_0}{\rho H}; \\ \dfrac{p - p^1}{\eta\rho H} - 1 \leq \dfrac{a^1{}_0}{A_1} \times \dfrac{L}{l^1{}_0} \times \dfrac{\pi - \rho h^1{}_0}{\rho H}; \end{cases}$$

$$\text{(III.)}\ldots\ldots \frac{S}{L} = 4\frac{H^1}{H}\Phi(p, e) = \frac{2V^2}{gH}\Phi(p, e);$$

where p denotes the pressure of the live steam, in atmospheres or any other units;

p^1, the pressure of the exhaust steam;

A, the area of the steam piston;

A^1, the area of the pump piston;

η, the ratio of the area of the pump piston to that of the steam piston, so that $A^1 = \eta A$;

e, the ratio of the cut-off to the whole stroke;

e_1, the ratio of compression to the whole stroke;

m, the ratio of the clearance volume to the volume of the whole piston stroke, so that the clearance $= mAS$;

ρ, the weight of one cubic unit of the liquid pumped;

a_0, the area of the pipe between the pump and the discharge air vessel at any section x, x_1;

h_0, the vertical height of the level in the reservoir above section x, x_1;

l_0, the length of the pipe between x, x_1 and the discharge air vessel;

$a^1{}_0$, $h^1{}_0$, $l^1{}_0$, the corresponding quantities for the suction air vessel and main (see sketch);

H, the resultant height through which the liquid is to be raised

being the total pressure height h, together with an additional height equivalent to the frictional resistance and the kinetic energy of the liquid;

L, the length of the column of liquid between the air vessels, together with the length of a column of liquid of cross-sectional area equal to that of the pump piston A^1 and of the same weight as the solid moving parts;

$\Phi(p, e)$, the function of the steam pressure and expansion degree given by equation (27).

Lengths, areas, pressures, and velocities may be expressed in any desired units, as the formulæ contain only ratios and not absolute quantities. To facilitate the use of these formulæ the following tables have been calculated, taking $p^1 = 1.1$ atmospheres (= 16.18 lbs.), $e_1 = 0.1$ and $m = 0.05$.

By means of these tables the application of equations I., II., and III. to the design and construction of direct-acting steam pumps and to the examination of those already in use is made easy; and although by reason of the absence of practical constants the values obtained from these equations must be looked on as approximate only, yet their application will show clearly in each case how the motion of the pump is to be governed, and these approximate values may even prove satisfactory enough for practical purposes.

VII.

To illustrate the significance of these equations and tables, and their practical application, we shall apply them to the well-known Worthington direct-acting steam pumps. With regard to these pumps it seems strange at first sight that *most* of them are being arranged without cut-off of the steam inlet, and that to obtain expansion of the steam they have to be provided with additional compound cylinders, while a cut-off of the steam inlet can generally be obtained by different means, which would especially raise the efficiency of these pumps, working as they do with large clearances and a short piston stroke. The equations and tables explain this. The average dimensions of these pumps are: piston stroke, about 10 inches; mean velocity of the piston, about 20 inches per second; ratio of the area of the liquid piston to that of the steam piston, between 7 and $\frac{1}{4}$. The discharge air vessel is placed over the pump. Let us suppose that in such a pump, without compound cylinders, a cut-off is arranged at one-third of the stroke, and let the steam pressure at inlet

TABLE I., VALUES OF $\frac{\eta\rho H}{p}$.

$\frac{e}{p}=$	$\frac{1}{2}$	0.4	$\frac{1}{3}$	0.3	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{7}$
4	0.563	0.488	0.426	0.392	0.333	0.266	0.216	0.177
5	0.622	0.517	0.485	0.450	0.392	0.325	0.274	0.235
6	0.661	0.586	0.524	0.489	0.431	0.364	0.313	0.275
7	0.689	0.614	0.552	0.517	0.458	0.392	0.341	0.302
8	0.709	0.635	0.573	0.538	0.479	0.412	0.362	0.323

TABLE II., VALUES OF $\frac{p-p^1}{\eta\rho H}-1$.

$\frac{e}{p}=$	$\frac{1}{2}$	0.4	$\frac{1}{3}$	0.3	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{7}$
4	0.287	0.484	0.701	0.851	1.177	1.726	2.361	3.096
5	0.255	0.426	0.609	0.733	0.992	1.404	1.845	2.312
6	0.236	0.387	0.559	0.669	0.897	1.246	1.607	1.974
7	0.224	0.373	0.528	0.630	0.838	1.153	1.470	1.786
8	0.216	0.359	0.506	0.603	0.799	1.091	1.382	1.667

TABLE III., VALUES OF $\Phi(p, e)$.

$\frac{e}{p}=$	$\frac{1}{2}$	0.4	$\frac{1}{3}$	0.3	$\frac{1}{4}$	$\frac{1}{5}$	$\frac{1}{6}$	$\frac{1}{7}$
4	4.992	3.359	2.568	2.250	1.802	1.390	1.119	0.926
5	5.703	3.874	3.005	2.663	2.182	1.748	1.468	1.273
6	6.211	4.240	3.314	2.953	2.450	3.003	1.715	1.521
	6.588	4.512	3.542	3.171	2.647	2.191	1.902	1.706
8	6.889	4.730	3.721	3.334	2.804	2.340	2.045	1.849

be 5 atmospheres (= 73.53 lbs.). Then from Equation I. and Table I., supposing $\eta = \frac{1}{3}$ and $p = \rho \times 5 \times 33.9$, we have $\frac{\frac{1}{3} \times \rho \times H}{\rho \times 5 \times 33.9} =$ 0.485; *i. e.*, the possible raising height, including all frictions, etc., is $H = 247$ feet. The height of the liquid column from the pump piston to the surface of the liquid in the discharge air vessel, and from the pump piston to the surface of the liquid in the suction air vessel, if such exists, may be taken to be about 2 feet each. The length of a liquid column with a cross-sectional area equal to that of the pump piston, and of weight equal to that of the solid moving parts, can be easily and correctly determined for each special case. Suppose the thickness or height of the steam piston is 4 inches, and its specific gravity referred to the pumped liquid, in this case water, is 7.5: then the length of a column of water having a cross-sectional area equal to that of the pump piston, and of the same weight, will be $4 \times 7.5 \times 3$; *i. e.*, 7.5 feet. Suppose the weight of the other metallic parts, piston-rod, etc., be equal to a water column with the same base and 4.5 feet long: then $L = 2' + 2' + 7.5' + 4.5'$; *i. e.*, $L = 16'$. Hence from Equation (III.) and Table III., $\frac{\frac{10}{12}}{16} = \frac{2\,V^2}{32.2 \times 247} \times 3.00$; therefore, $V = 8.3$ feet. Thus in the case of a cut-off at one-third of the stroke, the mean piston velocity would be about 8.3 feet per second, in consequence of the large difference between the motive power and the resistances during the first half of the stroke. Such a piston velocity is of course out of the question, and hence it is found necessary in practice to give up the advantage of steam expansion in direct-acting steam pumps of average dimensions, where the employment of existing compensators for keeping the piston velocity moderate and uniform would lead to too great complication and increase in price of this cheap and simple engine; while by working these pumps with plain steam pressure during the whole stroke the surplus of the motive power over the resistances, and, therefore, also the piston velocity, can be regulated easily and as desired by the steam inlet valve.

Equation (III.) offers also sufficient means for keeping the piston velocity moderate and uniform, the steam being used expansively. These means are:

(1) Equation (III.) shows that the piston velocity is decreasing with decrease of H. Thus for $H = 11$ feet (the corresponding value of η is to be taken about $7\frac{1}{2}$) equation (III.) would give a mean

working velocity of about 20 inches per second. Therefore, in low service pumps a suitable working velocity can be attained even when the steam is worked expansively and no special arrangements for moderating the piston velocity are used.*

(2) Equation (III.) shows the piston velocity is decreasing with the diminution of the length of stroke S. This property of direct-acting steam pumps has, it seems, been already made use of in practice, as in almost all direct-acting steam pumps the stroke of piston is equal to, or even less, than its diameter.

(3) Equation (III.) shows that the piston velocity is less the longer is the liquid column enclosed between the air vessels; that is, the further the discharge vessel is placed from the pump. Thus, in the case of the above-mentioned pump, the value of L required to reduce the mean working velocity to 20 inches per second is, by Equation (III.) and Table III., given by $\frac{10}{L \times 12} = 2 \times \frac{(\frac{20}{12})^2}{32.2 \times 247} \times 3.00$; hence, $L = 398$ feet. Deduct from this: 12 feet, corresponding to the weight of the moving solid parts; 4 feet, the usual distance between the air vessels; and 10 feet, the supposed length of the suction of liquid (in the case of a long forcing column the suction air vessel can be dispensed with, as is shown later), and we see that, for the attainment of a mean working velocity of 20 inches per second, the forcing air vessel must be removed from its usual place to a distance of about 370 feet. The necessary condition for constant continuity between such a column of liquid and the pump piston will be, according to Equation (II.) and Table II., $\frac{a_0}{A^1} \geq \frac{374}{398} \times \frac{247}{34 + 237} \times 0.609$; *i.e.*, $\frac{a_0}{A_1} \geq 0.517$; and for the suction column, $\frac{a'_0}{A^1} \geq \frac{12}{398} \times \frac{247}{33.9 - 10} \times 0.609$; *i.e.*, $\frac{a'_0}{A} \geq 0.189$. Therefore, to secure stability of motion, it is sufficient to have the pipes between the pump and the discharge and suction air vessels of cross-sectional area not less than $0.517A^1$ and $0.189A^1$ respectively.

(4) As L includes not only the length of the liquid column con-

* In 1889 a London firm erected at St. Petersburg five duplex direct-acting compound condensing steam pumps for raising water from the Neva to filters placed in the ground, each pump having an output of about 8,000,000 gallons per 24 hours. In these pumps the ratio η is about 7, the final steam expansion 8, the boiler pressure 75 lbs., and the mean piston velocity about 20 inches per second without compensators.

nected with the piston, but also the length equivalent to the weight of the moving solid parts, by increasing the weight of these parts, or by the introduction of a special solid mass connected in any way to the piston rod, we can obtain a corresponding decrease in the mean working velocity. Such increase of L without increase of the distance of the forcing air vessel from the pump, besides diminishing the mean velocity, will also increase the stability of the motion of all the parts, making the connection of the liquid column between the air vessels to the pump piston more secure, as is seen from Equation (II.).

(5) From Equation (III.) it is seen that the mean working velocity is decreasing with increase of $\Phi(p, e)$, and from Table III. we see that $\Phi(p, e)$ increases with increase of the original steam pressure and with increase of the steam inlet (e). Suppose a direct-acting steam pump is constructed for a given working velocity and with a certain steam expansion degree. We can, leaving the quantities $S\,L\,H$ and η unchanged, vary the working piston velocity by altering the steam pressure (p) and the steam expansion degree (e); because Equation (I.) can be satisfied by different sets of values of p and e, while the value of $\Phi(p, e)$ changes. Thus, Table I. shows that for $p = 4$ atmospheres and $e = \frac{1}{2} - \frac{\eta \rho H}{p}$ has nearly the same value as for $p = 5$ atmospheres and $e = 0.4$, while Table III. shows that for the same value, $\Phi(p, e)$ decreases from 5 to 3.87. Thus the piston velocity can be varied within considerable limits, because, by means of the cut-off mechanism. the steam expansion can be varied and the steam pressure can always be varied as desired. Hence, in every direct-acting steam pump working under given conditions $S\,L\,H$ and η, and provided with a cut-off mechanism which has a scale showing different cut-off degrees, there can be marked on that scale the different mean working velocities corresponding to each cut-off degree; so that in setting a certain cut-off, and regulating the steam pressure to give the fixed length of stroke S, the engineer will at the same time obtain nearly the desired piston velocity marked on the scale as corresponding to that cut-off.

(6) Formula (II.) shows that in direct-acting steam pumps the influence upon the motion stability lies not in the absolute weight L of all the moving masses, but in the ratio $\frac{L}{l_0}$ of the total weight to the weight of the single water column enclosed between the

pump piston and the next air vessel, while the working velocity, according to Equation (III.), depends on the total weight L; consequently the pump motion can be made steady independently of its working velocity. Under the term, stability of motion in pumps, as far as the governing of the moving water columns is concerned, should be understood the condition that the three moving bodies—namely, the pump piston and the two water columns between it and the air vessels—should move together like one solid mass; *i. e.*, these water columns should always stick to the piston, begin to move and come to rest at the same time, because in such a condition no water blow can take place, and the opening and closing of the suction and discharge valves, if properly constructed, would occur in the right time. If such a motion stability can be safely attained at velocities twice or thrice as high as compared with those of the present existing pumps, and Formula (II.) shows that it is attainable, then why not use higher velocities and lessen the size and number of pumps? At higher velocities the total length L, as determined from Formula (III.), will be 3 — 4 times less, so that a short removal of the discharge air vessel may be enough to create a steady motion and a reasonable working velocity in accordance with Formula (III.), even working with considerable steam expansion.

The foregoing leads to the conclusion that in direct-acting steam pumps the moving masses offer in themselves sufficient means for the governing of their motion, and there seems to be no necessity for the introduction of artificial means for this purpose, such, for instance, as fly-wheels and special compensators. Of these natural means the most radical appears to be the utilization of a part of the pumped liquid to obtain any desired working velocity in a pump working under given conditions SHp and e. This mean used separately or in combination with some other of the above-mentioned means seems to be, according to Equations (II.) and (III.), so sure and, in comparison with existing compensators, so simple that its application may prove most suitable in practice, especially in cases of water and oil supplies at large distances, and even in cases where there is no liquid column sufficiently long it may be found of advantage to lengthen the column with the above end in view.

The above conclusions, of course, are made from a theoretical point of view, and there may arise many constructional difficulties in carrying out the system in practice. One especially suggests

itself: How could the engineer control the discharge air vessel which may be placed some hundred feet distant? As engineering obstacles are getting removed, little by little, this one and many others also may and, probably, soon will be overcome. The author has, indeed, already devised an arrangement in which most of the difficulties appear to be fully overcome.

DISCUSSION.

Prof. John E. Sweet.—It seems to the writer that the fundamental notion, mathematically demonstrated in the author's contribution, is one which can be understood and considered mechanically without the mathematical demonstrations. In fact, it is likely that such a consideration comes first, and must have come first, in the author's mind; but, being a man with mathematical training, he has taken that method to set forth his notion, overlooking the fact that a good many of us who build engines and pumps do not absorb ideas readily in that way.

The proposition, it seems, is simply this: to use what Mr. Chas. T. Porter called the reciprocating fly-wheel, and apply it to a direct-acting steam pump for the purpose of using the steam expansively. That is, he proposes to make a steam pump with an enormously heavy piston use a larger steam cylinder or higher steam pressure, not only to drive the water, but to put this heavy piston in motion; then to cut off the steam at some early point, using the inertia of the piston with the expanding steam to complete the stroke. The striking difference between Mr. Porter's application and that of the author is, that Mr. Porter used cast iron and the author uses water for the heavy element. Just how heavy the piston, if of iron, needs to be to meet certain assumed conditions is, of course, a mathematical problem, and one which figures can be depended upon to solve; but with the piston made of water another element comes in, which is, the water piston is one which can be very easily pulled in pieces, comparable to a cast-iron one made of disks held together by atmospheric pressure. As an example, assume we have a pump piston made up of disks of cast iron perfectly fitted together, with the piston-rod attached to the centre disk. It is easily understood that if we attempt to accelerate the piston too fast the rear disks would not keep up, and also at the last half of the stroke the forward disks m have received such an impetus that the piston would not

From this reasoning, it would appear there is a limit to the speed at which a water piston can be used. It would also appear that the water piston had better be largely if not wholly on the discharge side.

The employment of this system implies an action similar to that of a fly-wheel pump, that is, acceleration and retardation; and that, too, from rest to high velocity and then to rest, or nearly so, of all the volume of water between the air chambers, while the modern idea favors a continuous flow. The difference is that between a simple pump with pulsating action, or (so far as present invention has gone) less economy or more complicated machinery with continuous flow. The plan suggested by the author to one who is not a pump builder would seem worthy of an extended trial. And more than a single trial may be necessary to determine the best practical proportions. I believe there are too many unknown quantities for mathematics to wrestle with successfully at the outset.

Prof. D. S. Jacobus.—I have looked over the formulæ, and what impresses me most is that, leaving aside the technical difficulty of applying them as formulæ, we would have great trouble with the frictional resistances. It is plain that the frictional resistances are represented by a certain factor called R. Now the way his formulæ is made, it shows that this R includes all of the friction of the movable parts, together with the friction of the water in passing through the valves. If we mean to run the pump by high-pressure steam, of course we increase the speed. There is no factor to show how much that would be increased. The formulæ are useful, to my mind, in showing the possibilities and the limitations of such an endeavor, which they do in a very thorough way.

*Mr. David Guelbaum.**—I am much impressed by the broad view Professor Sweet takes of the main question at issue, viz., the employment of a part of the moving water column as a compensating medium in working the steam expansively, and I find it almost impossible to pass any of the professor's remarks without serious comment.

First, the analogy of the water column enclosed between the air-vessels with a fly-wheel or a solid, heavy piston is true only as far as its main effect is concerned, *i. e.*, retarding or accelerat-

* Author's closure, under the Rules.

ing motion; but their practical working will be radically different. Indeed, for employing a water column as compensator you need only cut the discharge main in two parts, by removing the air-vessel from near the pump and putting it further away on the main. As the water pipes should always be and usually are full of water, the mean velocity of the water column serving as fly-wheel must remain the same as before, therefore the loss by friction also will remain nearly the same, whether we employ the water for compensating purposes or not. Quite different is the case with a fly-wheel, a heavy, solid piston, or any other compensator; here you will have to face the expense of increased friction and wear, and, besides, pay for the extra weight and mechanism. Further, if the boilers are priming, or the steam-pipes carry much condensation water, many break-downs do occur, even in the most perfect fly-wheel steam-engines; and the same must happen with a heavy, solid piston serving as fly-wheel. But nothing of this kind could happen in a direct-acting steam pump with a water column serving as compensator; at a sudden stop of the pistons the water would continue for a while to move through the valves, setting quietly back upon the valves, supposing the latter are arranged as they ought to be.

Professor Sweet remarks further that the modern idea is for a continuous flow of water. If this means the superiority of rotary and centrifugal pumping, I beg to oppose this modern idea of superiority. It is a recognized principle that mechanical work applied directly to overcome resistances is the most economical work. In forcing water through a main by means of a direct-acting steam-pump, placed between two air-vessels, we apply the power in the most natural way, and can have the flow of water continuous to any desired degree. I consider the air-vessel the most simple and rational means for getting a desired continuous flow of water; but the fact is, that the principle and function of the air-vessel are hardly understood thoroughly, and its arrangement is frequently defective even in the best pumping plants.

Concerning the professor's opening remark, that the fundamental notions of the exposed theory could be mechanically understood without mathematical demonstrations, I could agree with this view as far as the mere conception of the idea of employing the moving water as compensator is concerned. But it is when you start to realize in practice the abstract idea, that the services of mathematics become indispensable as a true guide.

UofM

Imagine only the amount of work that would be necessary to go through, in experimenting and manipulating with a pumping plant having at least 500 feet of water mains, for ascertaining the existing connection in different combinations between the mean working velocity, water head, length of stroke, length and diameter of water mains, and the mode of steam distribution, as this is given in the final equation III.; or to find out the conditions under which the water, while serving as fly-wheel, would not *pull to pieces*, as given in equation II. I think the work to be performed with the above end in view may be found not only hard, but also endless; and yet these things would be useful to know for a builder of such pumping plants.

DLXII.*

A MODERN DISINFECTING PLANT.

BY W. H. FRANCIS, PHILADELPHIA, PA.
(Member of the Society.)

IT has been truly said that the mechanic handles "the lever which moves the world," for every advance in science, art, or civilization calls for his aid.

The surgeon may be skilled, but the mechanic makes that skill efficacious by producing necessary instruments. So, also, following the great discovery by Dr. Robert Koch of the germ theory of disease propagation, the mechanic was called upon to design apparatus to destroy the germs and prevent the spread of contagious disease. The late threatened cholera invasion has directed especial attention to this subject, and numerous disinfecting plants have been erected, more especially by the Government through the efficient Marine Hospital Service, of which Dr. Walter Wyman is the able Surgeon-General.

A brief description of the machinery of a modern station may be of interest to the Society.

There are three ready means of destroying bacilli—by HEAT, by FUMES, by CHEMICALS. The plant has therefore been provided with apparatus to apply these three means.

BY HEAT.

In 1883 Dr. Koch, in connection with Doctors Guffky and Löffler, discovered that wet steam of 212° Fahr. would destroy bacilli by extended exposure thereto, and this has since been verified in fuller detail by other scientific research. In application of the above, two rectangular steam-jacketed chambers 16 feet long have been provided, with steam-tight doors opening at each end. The chambers are constructed of an inner and outer steel shell 2½ inches apart, cast-iron end frames, intermediate truss bands, and of screw stay-bolt construction.

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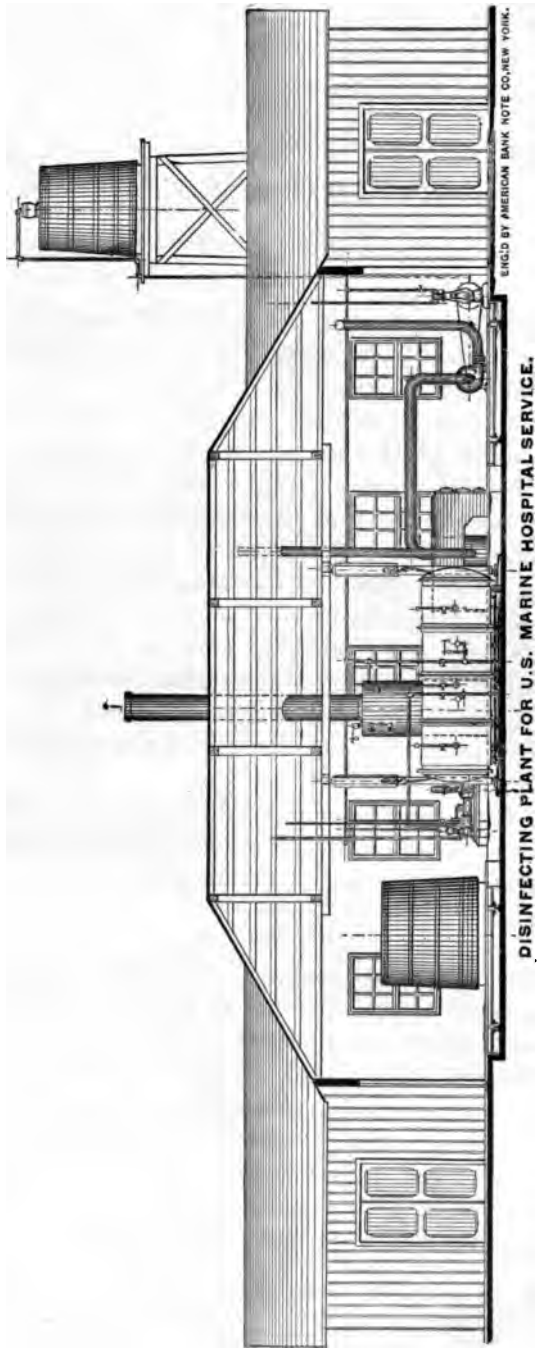
* Presented at the New York Meeting (December 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.



DISINFECTING PLANT FOR U.S. MARINE HOSPITAL SERVICE.

FIG. 17.

The doors have concave steel plates riveted to cast angle frames fitted with heavy rubber gaskets; they are handled by convenient cranes and drawn tight by drop-forged steel eye-bolts, swinging in and out of slots in the door frames. A number of chambers, notably at Charleston, Mobile, and San Francisco, are plain, cylindrical, and unjacketed, but this style is open to objection. The rectangular form was adopted in preference to round, as it gave the most effective space during exposure, with little loss of steam, and enabled cars on tracks to be readily handled in and out. The jacket was used to give perfect circulation and distribution of heat, to prevent too rapid condensation, and to dry the goods exposed. The jackets, which are filled with steam during the entire operation of the plant, make the chambers drying ovens, so that the articles to be disinfected are brought to temperature before the admission of steam to the inner chamber, and thoroughly dried after the steam has been exhausted.

In the experiments by Dr. Koch in connection with Dr. Wolffhugel, it was found that hot air alone, even at temperatures of from 230° to 248° Fahr., after an exposure of three hours, would not destroy the bacilli. It was necessary, therefore, to eliminate the possibility of air pocketing in the chambers, or even permitting a mixture of steam and air to remain during exposure. To prevent this, a vacuum pump was attached to the system of piping, whereby a vacuum of 15 to 20 inches was produced in the chamber before the admission of steam.

In all previous chambers, so far as the writer is informed, this important point was neglected and accounts for the reported unreliability of a number of plants.

For convenience of handling the goods, four cars have been provided, of light wrought-iron construction, with removable trays covered with galvanized screens, having a series of bronze wardrobe hooks in the top; permitting the articles to be laid out upon the trays, or, in the case of finer clothing, to be hung upon the hooks. The doors at both ends allow the cars to be brought in at one end and removed at the other, thus securing complete isolation of the infected and disinfected articles. After exposure, these cars, upon being unloaded, are returned to the working end of the chambers by means of transfer tables and side tracks, enabling a continuous alternating working of the plant.

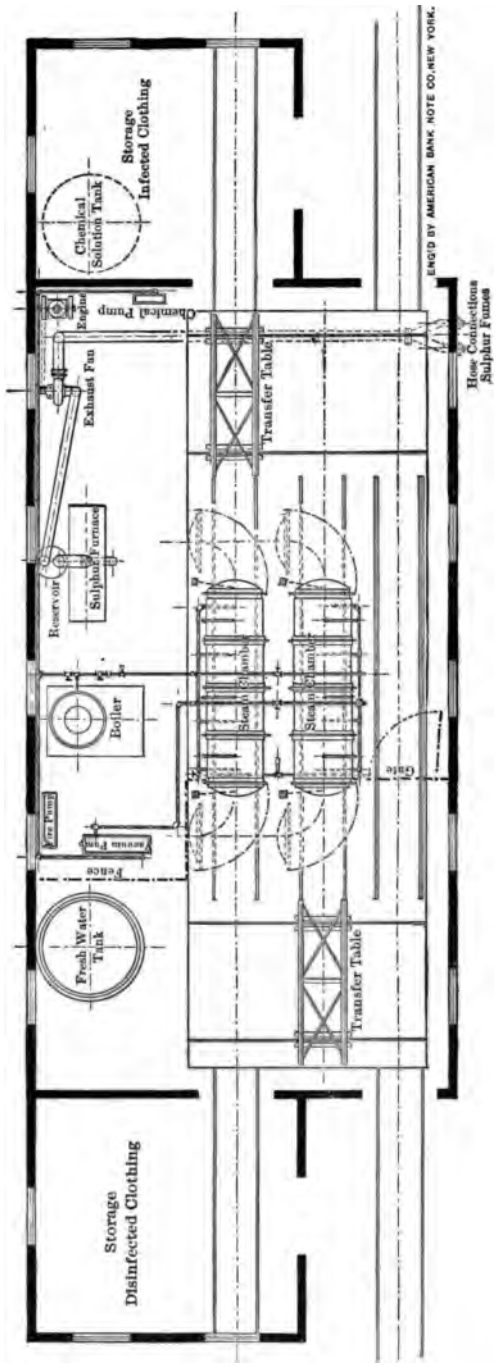

Storage
Infected Clothing
Chemical
Solution Tank
Engine
Chemical Pump
Exhaust Fan
Reservoir
Sulphur Furnace
Boiler
Fire Pump
Aqueous Pump
Fence
Fresh Water
Tank
Storage
Disinfected Clothing
Transfer Table
Steam Chamber
Steam Chamber
Transfer Table
Gate
How Connections
Sulphur Fumes
ENG'D BY AMERICAN BANK NOTE CO. NEW YORK.

FIG. 18.

As to the exact temperature required, the "doctors disagree." The consensus is from 100° C. (212° Fahr.) to 115° C. (239° Fahr.). This plant has therefore been arranged for any pressures up to, say, 15 lbs. (250° Fahr.), if required, the steam being generated in the boiler at high pressure, and the reducing valve set to the pressure required.

The system of piping is so arranged that steam may be admitted to the bottom or top of the chambers at will, through several openings, and have perfect circulation. Galvanized hoods are placed in the chambers, so that steam is not forced directly on the clothing. Each chamber is provided with thermometers to register the temperature, vacuum and steam gauges, safety valves, traps, etc., and is covered with magnesia non-conducting covering.

BY FUMES.

Fumigation is most generally by sulphur dioxide gas, whose action is exerted as sulphurous acid. These fumes are produced by burning sulphur in the air, say 3 lbs. to every 1,000 cubic feet of air space. A crude way is to place sulphur in iron pots, pour alcohol over it, set fire thereto, and close all openings. This gives no control of operation nor assurance of thorough fumigation.

In this plant, a double furnace has been provided with small fire boxes at each end, over which are two cast-iron shallow basins, about 5 feet long each, the whole enclosed in a sheet-iron frame. The sulphur is placed upon the basins and the fire lighted, quickly melting the sulphur, which soon ignites. To prevent too rapid combustion, baffle plates are arranged, and the proper quantity of air admitted through adjustable valves in the furnace fronts. The fumes thus generated pass to a reservoir, from which they are sucked by an exhaust fan, and are thence forced through piping and large flexible asbestos hose to the apartment to be fumigated.

BY CHEMICALS IN SOLUTION.

The chemical agents commonly used are solutions of bichloride of mercury (corrosive sublimate), carbolic acid, and chloride of lime.

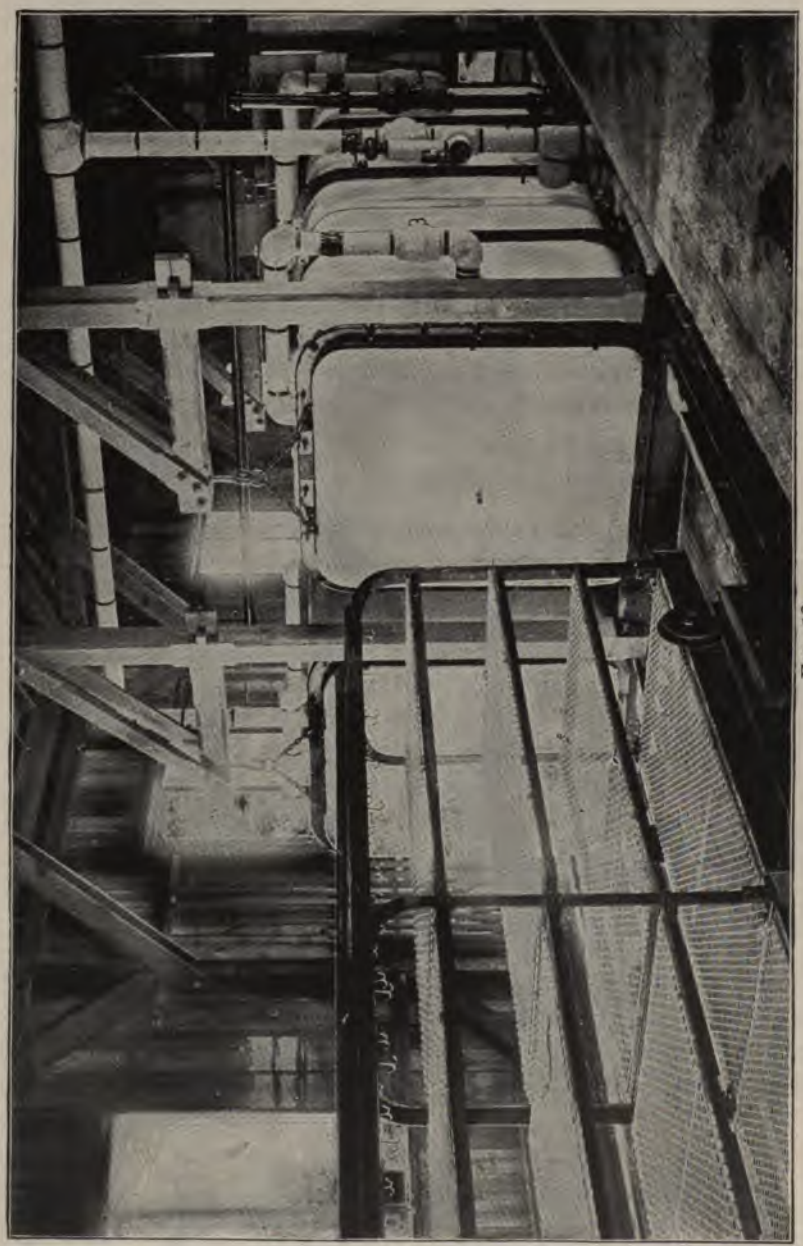


FIG. 19.

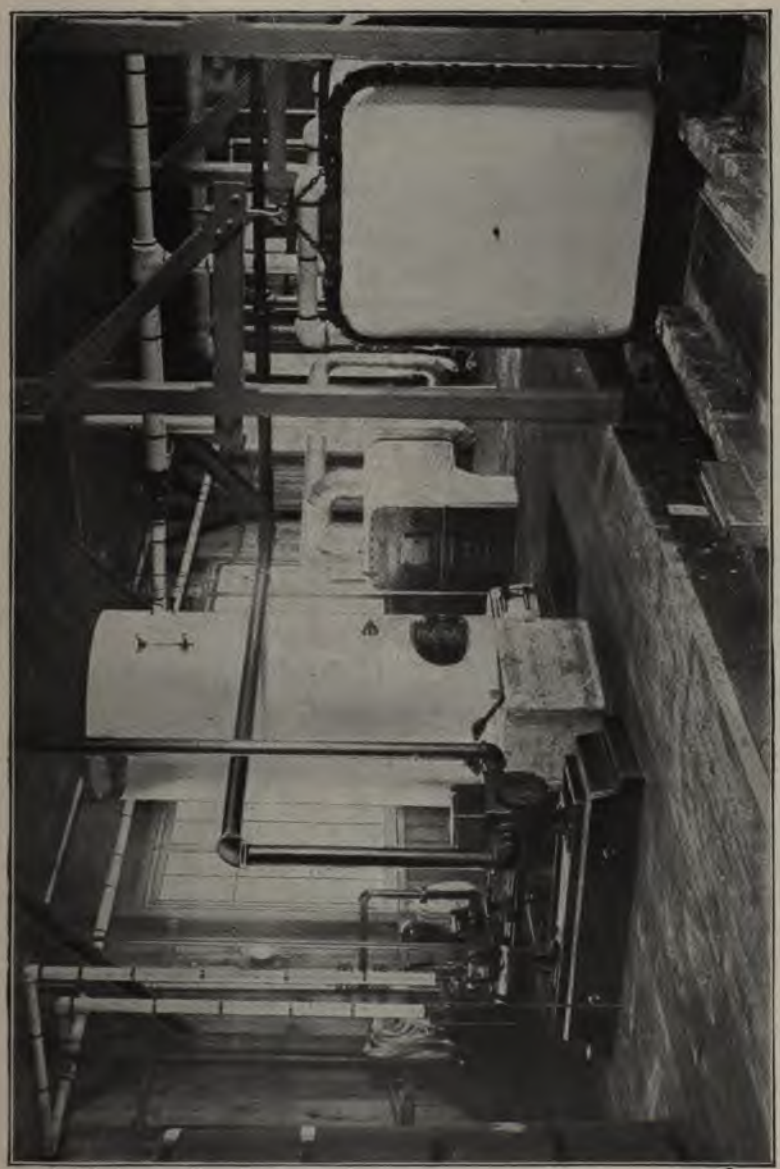


FIG. 20.

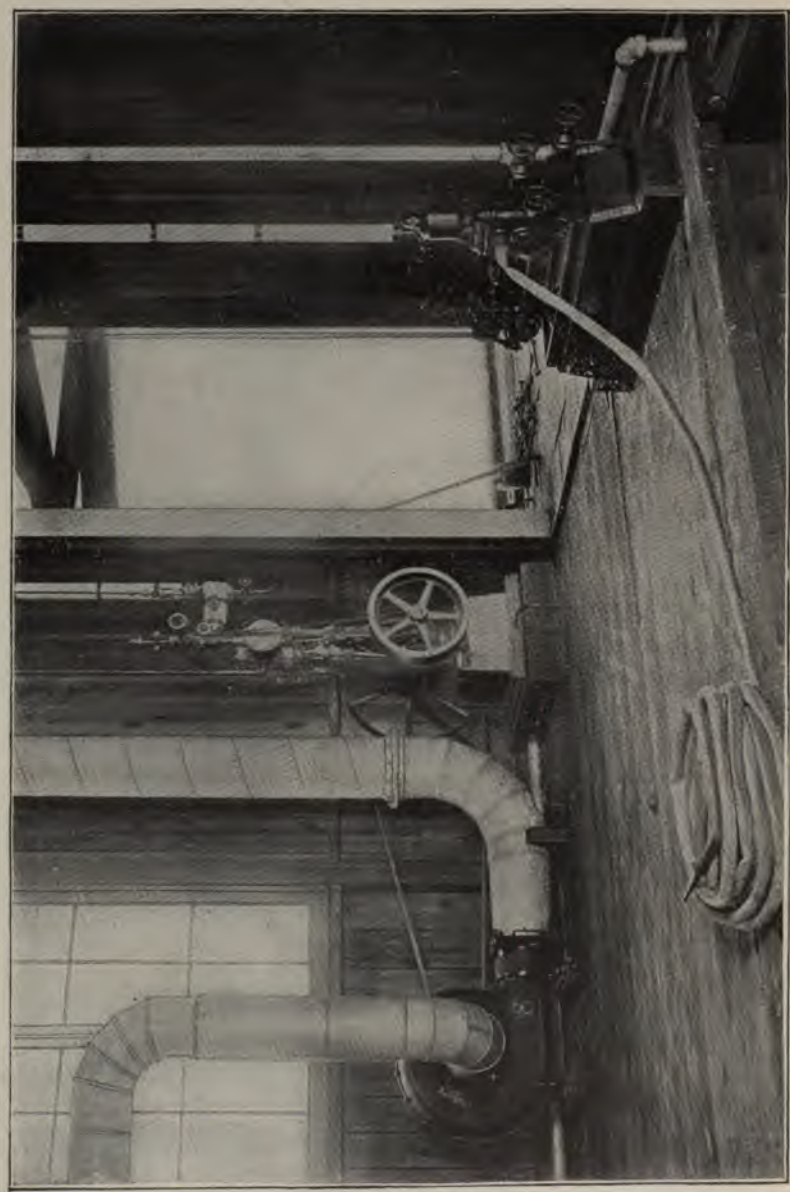


FIG. 21.

The first named is most generally used, in strength, say, 1 to 1,000, and the mechanism required is very simple. A large wooden tank is provided with a steam pump piped thereto. The chemicals are placed in a heavy keg perforated with numerous small holes, arranged at the top of the tank, and the water in sufficient quantity pumped into it slowly, making the solution of required strength. This solution is then pumped through one-inch hose under pressure, insuring penetration into cracks and corners of the apartment being disinfected.

The plan and views from the station will illustrate the above brief description (Figs. 17 to 21).

DISCUSSION.

Mr. O. C. Woolson.—I am not able to gather from the author which one of these three processes he contends is the best one, but he starts off with his scheme of steam disinfecting apparatus, and possibly it may be proper to assume that he thinks that this is the best. But from his description I should, from my slight knowledge of the business, be inclined to doubt its thorough efficacy unless some system of repeating is in vogue. Simply treating the germs with one dose of steam, of any temperature, and considering that as sufficient, I am inclined to doubt very seriously. I am strongly of opinion that two applications would be scarcely enough. I have been told by good authority that it requires at least three applications. The first application will kill a large percentage of the germs, and at the same time it will propagate another percentage, which would not become apparent until that application had been gone through with and the steam had been drawn off and everything cooled down, and then they become active; and after another application those that have become active may be destroyed, and yet in that application it will propagate a certain other percentage which can only be destroyed by still another application; and I am of opinion now that it takes at least three to destroy the germs entirely. If the idea is that one application of this steam process is sufficient, I think it is an error.

Mr. W. B. Le Van.—I would state that in relation to germ killing we were troubled with moths in some furniture, and we sent it to an establishment in Philadelphia which put it into a room where

it remained for forty-eight hours under steam pressure of twenty pounds to the square inch, 260°. After the furniture was returned we found nearly as many moths as before the application. It was sent back and had a second forty-eight hours' application, and then we found a considerable number of moths flying from the same furniture after ninety-six hours test. So that shows that the gentleman is right in saying that one simple application does not kill germs of that character.

Mr. Wm. H. Francis. *—Mr. Woolson seems to have overlooked that the object of this paper is not to present theoretical ideas of disinfection, but to describe the actual, practical application of scientific data up to date, in a plant now in successful operation.

His first query as to the best means of disinfection, of the three noted, is one of adaptability entirely. For instance, in the case of a vessel, the hold and its cargo might readily be disinfected with sulphur fumes, where steam could not be applied; and bedding and clothing readily steamed without damage, when fumes or chemicals would destroy not only the germs but the articles also. Each means is equally efficacious when applied with judgment as to its adaptability.

From all authorities consulted and from the reported results of steam disinfection, more especially in the case of the "comma bacillus" of cholera, I must disagree with Mr. Woolson's supposed necessity for several applications. The viability of the germ is very slight. At ordinary temperatures, not below 57° Fahr., it grows, and while lower temperatures retard growth, they are only destroyed by the higher temperatures above even 132° Fahr., depending upon the surroundings.

An exposure to sunlight for five hours has been shown to be sufficient entirely to destroy the vitality, under certain conditions. All data point to the fact that higher temperatures do not propagate, but destroy, the germs.

I note the results of some experiments on the viability of the cholera germ, at the ordinary temperature of the room, say 70° Fahr. The vitality of bacilli entirely disappeared, as follows:

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* Author's closure, under the Rules.

Sour cherries, 3 hours;
Sweet cherries, 3 to 7 days;
Fresh fruit, 5 hours;
Dried fruit, 1 to 2 days;
Oranges and bananas, 24 hours;
Beer, 3 hours;
White and red wine, 5 to 15 minutes;
Cider, 20 minutes;
Cold coffee, 2 hours;
Unsterilized milk, 24 hours;
Sterilized milk, 9 to 12 days;
Fresh fish, 2 days;
Salted and smoked fish, 24 hours;
Bread, exposed, 24 hours;
Bread, wrapped in paper, 3 days.

The example quoted by Mr. Le Van is not analogous, as heat is the vivifying influence of moths' eggs, as it is with ordinary hens' eggs.

In conclusion, it must be admitted that an element of uncertainty exists in all human agencies, perfect disinfection being total destruction of the articles by fire, but uncertainty is reduced to a minimum by the care and skill of an experienced operator.

DLXIII.*

USE OF THE INDICATOR FOR CONTINUOUS RECORDS IN DYNAMOMETRIC TESTING.

BY WM. S. ALDRICH, MORGANTOWN, W. VA.
(Member of the Society.)

FOR a variety of purposes the steam-engine indicator is probably one of the most serviceable spring-pressure-recording instruments at the command of the engineer. It has reached a high degree of mechanical perfection for its particular use in steam-engine testing. Notwithstanding its well-known defects when applied to this purpose, it seems only possible to perfect further its mechanical construction, so long as its typical piston mechanism is adhered to. Still, this is not an inherent objection to the use of the indicator in some other lines of recording work, and the purpose of this paper is to show how it has been so applied in power testing with a dynamometer. It is sometimes better to make use of an instrument which has been brought to a high degree of perfection in one field of work than to undertake the development of new and untried forms of apparatus for limited and usually special tests. Taking the indicator, therefore, as we find it, and not delaying work in which it may be used to advantage till it (or a similar instrument) becomes what it ought to be, the steam-engine indicator will be found extremely serviceable in obtaining short continuous records in power tests.

Almost all tests of this character may be reduced to the measurement of a more or less variable pressure, depending on the amount of energy being absorbed by the dynamometer. This pressure is usually measured as a pressure on a platform scale, or by the maintenance of a weighted lever in a position of equilibrium, or by the simple pull on a spring balance. But this pressure may be transferred or transmitted to any other instrument designed for recording variable pressures, as the indicator, for instance. In some cases, simple mechanical

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

connection may be used, while in others, hydraulic connection will be found desirable.

Mechanically transmitting the pressure from the arm of the dynamometer, by a linked connection, to the indicator piston, enables us to turn the indicator into a recording spring balance. In this method its piston mechanism is not in the least objectionable, serving merely as a guiding mechanism.

In the transfer of the pressure from the dynamometer arm to a diaphragm, which in turn presses upon water, oil, or mercury, in a closed vessel, and thence, by pipe or tubing, to the cylinder of the indicator, we make use of it as a recording pressure gauge. Small variations in pressure, slowly increasing or decreasing, yet of comparatively short duration, are not faithfully recorded on account of the retarding effect of friction of the indicator piston; but the leakage past the piston will be the least. Rapid and extreme variations in pressure, as well as extreme pressures, maintained for any length of time, are sure to result in more extended leakage, even when mercury is used under the indicator piston, as arranged for the form of the "U" manometer. When the greatest sensitiveness of the indicator is most desired (with small pressure variations) it is least obtained. It would therefore be much better to use some form of diaphragm, or closed-tube pressure-recording gauge. It should be capable of enduring much more rapid and extreme fluctuations than any on the market at present.

The requirement of an absolutely closed system, under all conditions of use of the hydraulic connection, corresponds to the necessity for inextensible (or rigid) parts or members of a linked connection. In either case, there must be an invariable relation, or fixed proportionate transmission of motion, from the dynamometer lever arm to the indicator piston. The extent of movement of this piston must correspond exactly to the pressure which produced that movement, while the piston is reacted upon by the indicator spring.

The time interval must be short, during which the power absorbed by the dynamometer is to be measured and recorded by the indicator, as at present constructed. For tests of from one to two hours' duration (the usual laboratory limit) a larger drum might be substituted; that is, a drum with as large a diameter as possible, consistent with the present distance between the axis of the drum and the indicator cylinder. It is under-

stood that the drum is only turned once during the time interval, as at present in engine testing.

Turning the drum by leading the cord (better, stranded German-silver wire) to the leading screw of a shop lathe will be found quite satisfactory for almost all cases of shop and laboratory testing. It is true that the shop engine is required to do a variety of work, and whether it turn the grindstone or the drum of an indicator, its more modern type is reliable. With steady steam pressure, and shop work to perform which is only slightly variable, it will be found more readily and scarcely less satisfactorily harnessed to the indicator drum than a fine astronomical chronographic movement. Each, of course, has its field of usefulness. Work should not be neglected because the time intervals involved cannot be measured absolutely to within the one-hundredth of a second. Moreover, it will be found that the possible changes on an ordinary screw-cutting lathe are as extensive as may be desired for almost any kind of time movement. They are so much more varied, in fact, that I have found no maker of chronographs as yet anxious to furnish an instrument with one-tenth of the possible ranges of speed of which the shop lathe is hourly capable. The cylinders of the finest chronographs are back-geared to turn once in ten seconds and one minute; or, once in thirty seconds and one minute.

In passing, it may be well to note the necessity for a light-powered uniform mechanical movement, for general shop and laboratory testing, for other timing besides that as above noted in connection with the indicator. It appears that the change-gear system of an ordinary screw-cutting lathe has been so well developed and is so admirably adapted to a wide range of speed changes that all that is further required is to gear on such a system, or its like, to a small motor; and this motor should be preferably driven by a weight, and its speed controlled, as in chronographic movements, mechanically, by a delicate friction governor, or by a gravity or electric-contact arm.

It should be borne in mind that besides the above quantitative records, others, purely qualitative in character, may be desirable. For this purpose it is amply sufficient to draw the cord attached to the indicator drum along by hand. In this way most interesting, and it may be valuable, information can be recorded respecting the performance of a dynamometer; or some characteristic of its load variations examined into, without

requiring to find the mean pressure exerted by the dynamometer arm, or the mean load. Such characteristic variations may be due to changed conditions of the rubbing surfaces, in friction dynamometers, or to speed changes, or both. The records from the use of hydraulic connection are more of a qualitative value, as showing the fluctuations of power absorbed, and the time of occurrence of such, owing to the errors in the recorded pressure arising from leakage past the indicator piston.

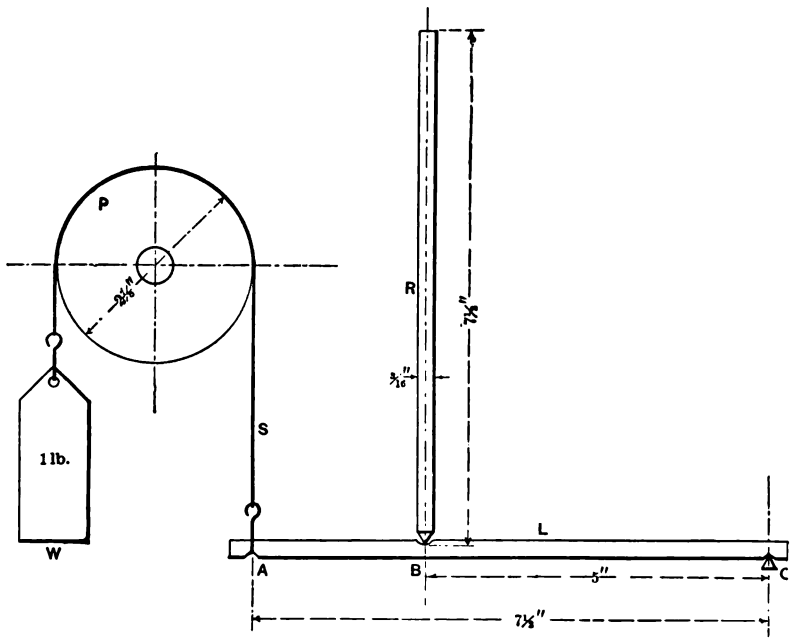

FIG. 22.

Arrangement of parts transmitting the pull of strap, *S*, to the rod, *R*, which presses against the indicator piston.

The timed records are integrated, by the planimeter, in the usual way, for the value of the mean ordinate, in this case, the mean pressure recorded. This corresponds to a certain mean pressure given off by the dynamometer arm, as found by plotting a curve of calibration of pressures, connecting those of the dynamometer arm with those of the indicator spring. In this way the mean or average pounds of effective pull on the dynamometer strap or other friction-absorbing device becomes known, from which the power absorbed is computed as usual.

I. MECHANICAL TRANSMISSION OF THE DYNAMOMETRIC PRESSURE.

In testing a No. 1, 6-inch, Pelton water-wheel, absorbing only a fraction of a horse-power, at high speed, the arrangement shown diagrammatically in Fig. 22 was adopted. The light leather strap, *S*, passed over the motor pulley, *P*, and was attached at the lower end, *A*, to the lever arm, *L*, while the other end supported a counterweight, *W*, of 1 lb. The lever moved upon a knife edge, at *C*, and pressed the rod, *R*, firmly against the

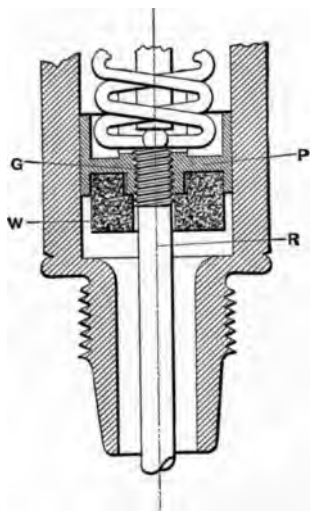

FIG. 23.

Arrangement of rod, *R*, simply pressing against the indicator piston, *P*.

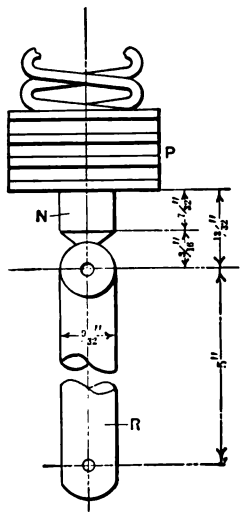

FIG. 24.

Linked connection of rod, *R*, to the indicator piston, *P*.

piston of the indicator. Fig. 23 shows the arrangement of the upper end of this rod, *R*, pressing directly against the piston screw, *G*, and inclosed by a rubber washer, *W*. Fig. 24 shows a linked connection, designed and specially made for this purpose, by the Crosby Steam Gauge & Valve Co. The piece, *N*, is screwed into the piston, after the regular screw, *G*, is removed.

In making the tests, the indicator cord was attached to the tool-post of a lathe, and driven by the leading screw, the change gears being arranged for any definite time movement, as may be desired: thus, for a three-minute interval, as shown in Figs. 25 to 30, inclusive. After pulling the cord taut, the time

interval was noted either by marking the card and noting the time, or by touching the lever (or other part of the system, as outlined in Fig. 22), and thus causing the indicator pencil to make its own time record. In this way several time cards were taken and compared. The speed of the leading screw was also

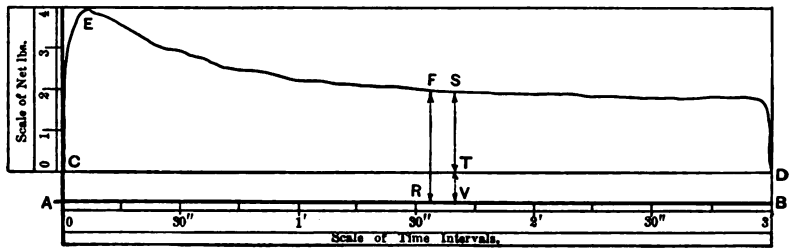

FIG. 25.

Card showing starting and stopping variations.

noted, from which, and its known pitch, the time interval for any combination was readily obtained, and the time base of the cards drawn, as shown. The pressure scale was determined by using a standardized spring balance, reading to one-fiftieth of a pound, and plotting the pencil movement corresponding to equal incre-

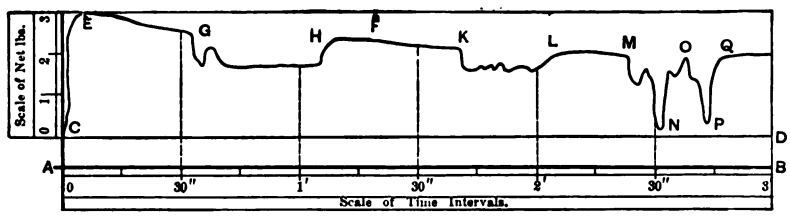

FIG. 26.

Card showing running variations, under average and extreme conditions.

ments of weight, as 1 lb., by attaching the hook of the spring balance to the hook at the end, A, of the lever L, the weight, W, remaining on, as during the subsequent tests. By this means, the net or effective dynamometric pressure (a pull) was recorded at once, and the scale of pressures drawn according to this calibration. On the cards, the line AB is the indicator zero,

found by relieving the rod, *R*, of any pressure upon it, and taken either while the motor is at rest, or running; the line *CD* is the dynamometer zero, taken while the motor is at rest, or drawn according to the previous calibration for this line, as compared with the indicator zero.

Fig. 25 was taken with oiled graphite on the strap, and shows

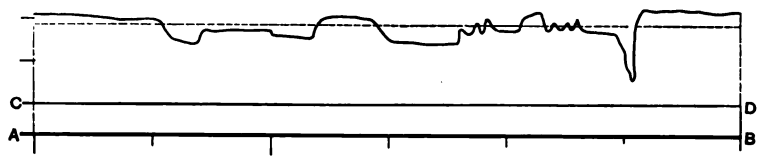

FIG. 27.

Three-minute power card. Mean speed, 1,366 revolutions per minute. Mean effective dynamometric pressure, 1.87 lbs. Dynamometric horse-power, 0.058.

characteristic variations, due to starting with a cold dynamometer pulley, and of stopping after it had been much heated by friction in running. The gradual fall of the line, from *E* to *F*, is caused by the decreased friction due to the heating of the pulley. The ordinate, *FR*, is the total pull on the strap; *TV* the pull due to the counterweight, *W*, and the other parts of system of leverage acting with it, amounting to 0.75 lb., on cards of Figs. 25 to 30, inclusive; and *ST* is the net or effective dynamometric pressure at the point considered.

Fig. 26 shows running variations, as follows: after starting,

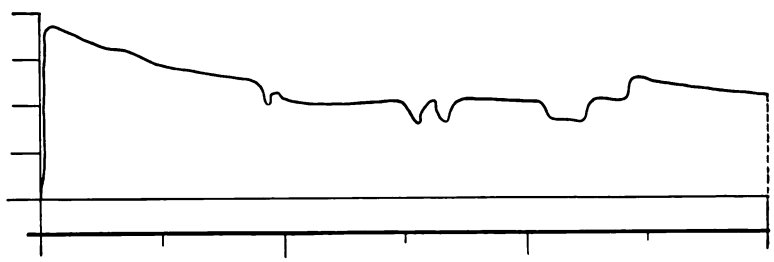

FIG. 28.

Three-minute card, showing starting and running variations of the dynamometric pressure, with dry powdered graphite on the strap.

the effective pull or pressure, due to the friction of the strap on the pulley, fell from *E* to *G*, as above noted; then a hydrant in the system of piping, branching off from the main water sup-

ply pipe leading to the motor, was slowly opened and closed, then left open to *H*; after closing the hydrant, the friction pressure fell, as usual, from *H* to *K*, while the pulley became more heated; the hydrant was opened at *K*, and closed and opened in quick succession, to *L*; *L* to *M* shows the usual effect of the

FIG. 29.

Three-minute card, showing starting and running variations under strap conditions similar to those of Fig. 28.

heating of the pulley; from *M* to *N*, the throttle valve (a hand-wheel gate-valve) was closed in part, opened, closed almost entirely, at *N*, opened again at *O*, then rapidly closed to *P*, and opened to *Q*, remaining open to the end of the card.

Fig. 27 is a power card. The area, above the line *CD*, as found by a planimeter, is 3.29 square inches, its length is 5.4

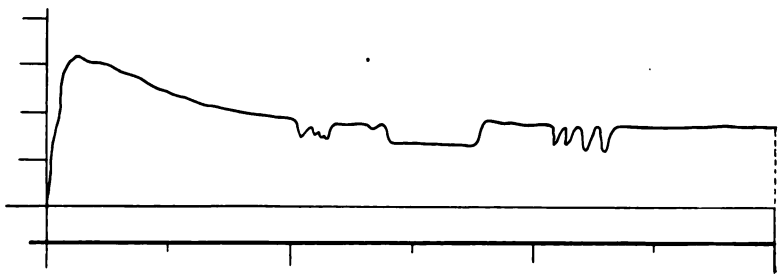

FIG. 30.

Three-minute card, showing starting and running variations, with oiled graphite on the strap.

inches, giving the mean ordinate of 0.609 inch; from the pressure scale, 0.325 inch corresponds to 1 lb.; therefore, the mean effective dynamometric pressure is 1.87 lbs. The total speed, as counted by a Tabor revolution counter, for the three

minutes of the run, was 4,100 revolutions, giving 1,366 revolutions per minute. The pulley has a mean diameter of $2\frac{7}{8}$

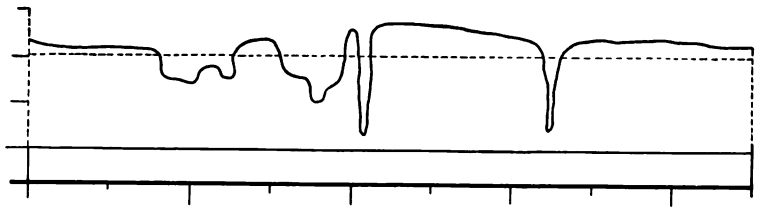

FIG. 31.

Four and one-half minute power card. Speed, 1,511 revolutions per minute. Mean effective dynamometric pressure, 2.04 lbs. Dynamometric horse-power, 0.0703. Taken with 10-lb. indicator spring, and leverage of 5 to $7\frac{1}{2}$ inches.

inches, from all of which the horse-power is computed, as follows:

$$\text{Dyn. H.P.} = \frac{\text{Mean Effec. Dyn. Pres.} \times \text{Circum. (feet)} \times \text{Mean Rpm}}{33{,}000}$$

$$= \frac{1.87 \times 0.7525 \times 1{,}366}{33{,}000} = 0.0582.$$

The mean dynamometric pressure is shown on this and subsequent power cards by a dotted line.

Figs. 28, 29, and 30 show characteristic starting and running variations, the former two cards, with dry graphite only on the strap, from cold condition of the pulley; and the last card with oiled graphite (powdered or flaked), also with cold pulley, at

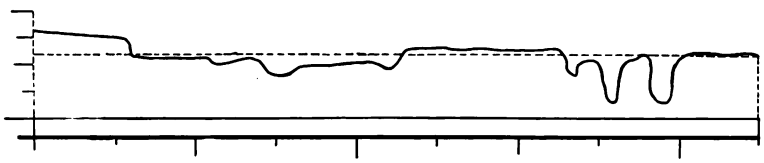

FIG. 32.

Four and one-half minute power card. Speed, 1,533 revolutions per minute. Mean effective dynamometric pressure, 2.4 lbs. Dynamometric horse-power, 0.0839. Taken with 20-lb. indicator spring, and leverage of $2\frac{1}{2}$ to 5 inches.

the start. The small variations in the curves are due to a hydrant being opened and closed, and the larger variations to the throttling of the water pressure, at the gate throttle valve.

Fig. 31, as well as all of the preceding cards, were taken with a 10-lb. indicator spring, acting at the leverage, and by the arrangement, as shown in Fig. 22, of 5 to 7½ inches. But Fig. 32 shows a card, taken with a 20-lb. indicator spring, and leverage of 2½ to 5 inches ; that is, the indicator rod, *R*, was midway between the fulcrum, *C*, and the other end, *A* ; and the Crosby link connection, Fig. 24, was used. The time interval of cards, Figs. 31 and 32, was four and one-half minutes, and the total speeds, respectively, 6,800 and 6,900 revolutions, from which,

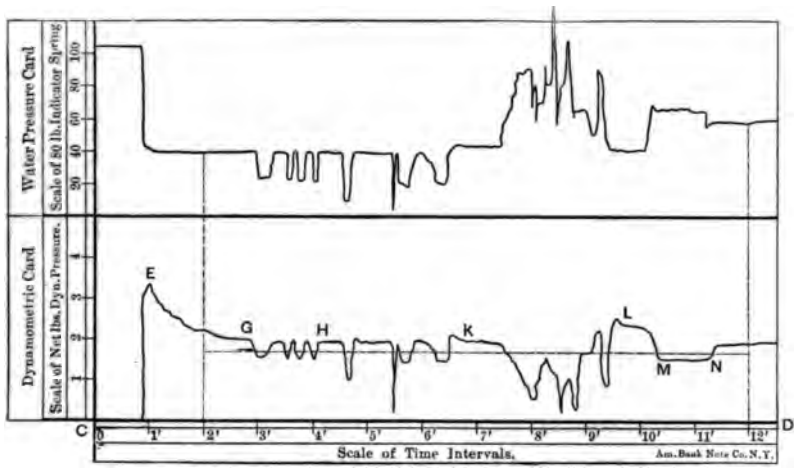

FIG. 33.

Simultaneous dynamometric and water-pressure cards of 6-inch Pelton Water Wheel. Mean speed, 1,210 revolutions per minute. Mean effective dynamometric pressure, 1.656 lbs. Dynamometric horse-power, 0.0457.

and the other data, obtained as previously explained, the dynamometric horse-power was computed.

Fig. 33 shows simultaneous dynamometric and water-pressure cards, taken for about twelve and one-half minutes, but only ten minutes of this period were used to find the mean speeds, pressures, and horse-power, beginning at the 2-minute ordinate, as shown, after the motor had settled down to almost steady running. The dynamometric card was taken with a 10-lb. spring in the indicator, using the Crosby link connection of Fig. 3, with the leverage of 5 inches to 7½ inches, as shown in Fig. 22. The strap was thoroughly covered with oiled powdered graphite. The water-pressure card, showing the starting and running

variations in the head of water, was taken with an 80-lb. indicator spring, the indicator being located between the gate throttle-valve of the motor and the main stop-valve of the line supplying the water, and its drum being drawn simultaneously with that of the other indicator. From *G* to *H*, the variations are due to opening and closing a hydrant in the system branching off the main supply pipe; from *H* to *K*, the main stop-valve of the pipe line was partly closed, and wholly closed for an in-

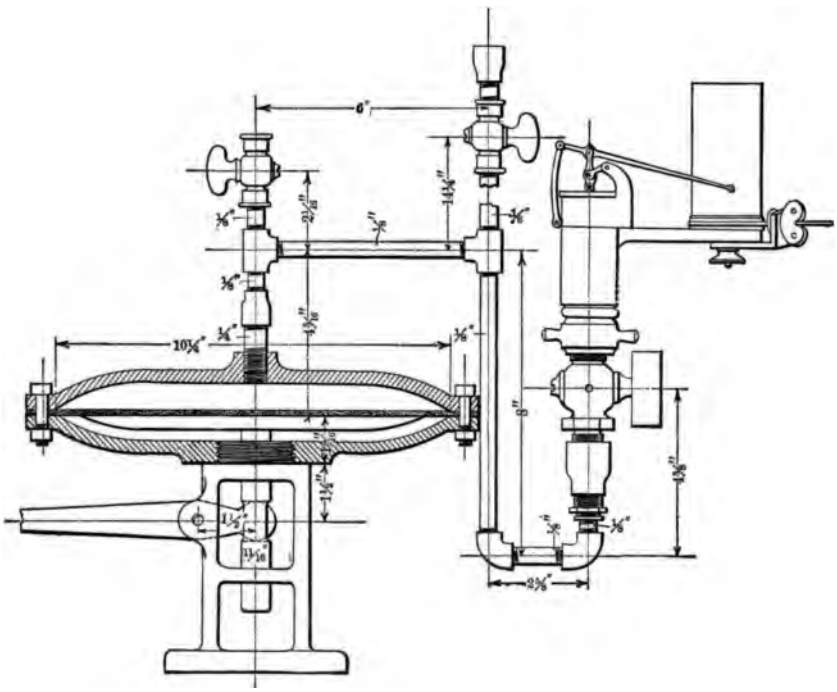

FIG. 34.

stant; from *K* to *L*, the motor's gate throttle-valve was partly closed, and wholly closed for an instant; from *M* to *N*, the motor was running with its own throttle-valve partly closed, being fully opened again at *N*. From the 2-minute to the 12-minute ordinate, the integrated area is 2.22 square inches, the length of this time base is 4.125 inches, giving the mean ordinate as 0.538 inch, or 1.656 lbs., on the vertical pressure scale of 0.325 inch to the pound dynamometric pressure. The total speed for the ten minutes was 12,100 revolutions, or a mean

speed of 1,210 revolutions per minute, giving the dynamometric horse-power of 0.0457.

II. HYDRAULIC TRANSMISSION OF THE DYNAMOMETRIC PRESSURE.

Fig. 34 shows the arrangement adopted in this case. The variable pressure to be measured and recorded was applied at the end of the lever arm, shown below the diaphragm casing. The pressure was thus transmitted to the diaphragm, through the vertical stem pressing against the enlarged plate beneath

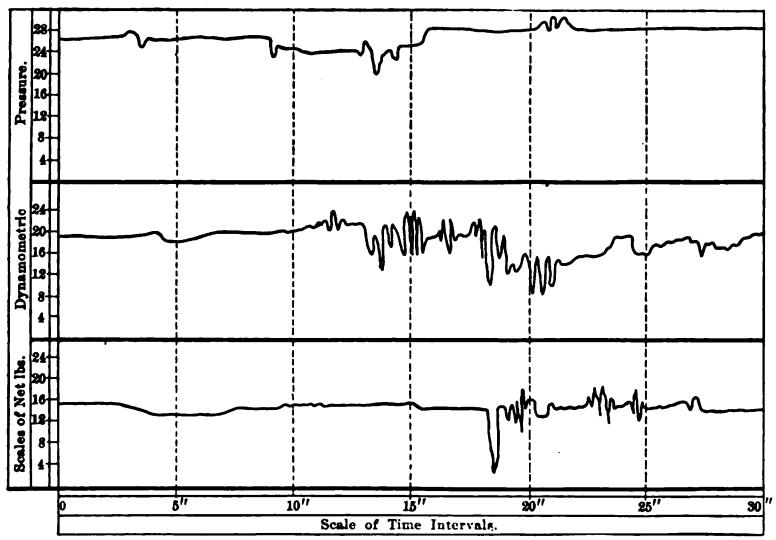

FIG. 35.

Three cards, on the same time base, showing variations of dynamometric pressure recorded by the indicator, arranged as in Fig. 34.

the diaphragm—a rubber disk about $\frac{1}{16}$ inch thick. The casing and system of piping above the diaphragm was filled with water, after the "U" tube had been filled with mercury up to the indicator piston. The cock directly over the diaphragm was for the escape of air while the system was being filled through the stand of pipe to the right. Before filling the "U" tube with mercury it was first filled with a high grade of oil, then the mercury poured down the pipe to the left of the indicator till it reached almost to the level of the horizontal branch leading to the diaphragm. In this way the "U" tube system, as well as

all parts of the indicator connection, became lined with the oil, which flowed out of the escape-holes of the indicator as the mercury slowly rose up to the same level. To put the indicator piston in position it was pressed down upon the mercury, which was allowed to escape through the small drain-hole in the indicator cock. In this manner an approximately closed system was obtained, as free as possible from air.

Fig. 35 shows three cards, arranged on the same time base, as taken by this method. The scale of pressures is the same in each case, and is taken from a calibration of the indicator, in this position, for 4-lb. increments applied at the end of the lever arm, shown below the diaphragm of Fig. 34. In running the pressure up above 28 lbs., for this calibration, the mercury oozed out of the indicator escape-holes as the lever arm was further pressed down to hold the pressure of calibration.

With an absolutely closed system, this method adapts itself to traction dynamometric work,* the pull (or push) being entirely by and through the lever arm above referred to, and thence transmitted to a diaphragm pressure-recording gauge placed inside the car. Of course, the chart of this pressure-recording gauge may be moved by belting to the car axle, or otherwise timing its movement with that of the car, as with the Boyer speed recorder. But it cannot be integrated for mean values, in this instance, unless the car speed remains absolutely uniform, which is never the case. Therefore it is best to drive the recording chart by means of clock mechanism, attached to the same, and placed with it inside the car. To prevent derangements due to starting suddenly and to unforeseen extreme variations, a safety-valve air-chamber may be introduced in the system of piping leading from the diaphragm to the indicator. This method of relief would become immediately operative, so soon as the pressure within the system exceeded the limit of the safety-valve.

As a machine-tool dynamometer, also, the method outlined last will be found useful in measuring and recording the pressure upon the work undergoing cutting operations, as in drill press work, the working pressure of which would be recorded by placing the work in direct connection with the lever arm of the dynamometer. If the pressure-gauge chart be moved by

* As has been thoroughly tested and practically developed by the Richard Frères, Paris, in their diaphragm (or springless) traction dynamometer.

connection to the automatic feed mechanism of the drill press, its timing will be regular enough to allow of integrating the record for the mean effort exerted during the drilling operation.

For the Alden, Prony, Webber, Lewis, and other forms of scale-arm absorption or transmission dynamometers, the method of Fig. 34 will be found equally adapted, by using a closed system with a diaphragm pressure-recording gauge.

DISCUSSION.

Prof. R. C. Carpenter.—I think it has been quite customary to use the indicator because of its convenience for many of the things which are mentioned in the paper, and I think we are indebted to Professor Aldrich for many ingenious suggestions which are here made. There is just one thing in regard to the use of the indicator on a "closed circuit" in which my experience has not been the same as reported here; in fact, it has been quite different from that. We have tried for the last three years, in a number of investigations, to use the indicator on a "closed circuit" connected with a dynamometer, in a manner similar to that described in the latter part of this paper, and in every single case we found that the errors were exceedingly large. In short, we found that either the indicator would leak or the piston would stick. Now, in a closed circuit, we have a very limited amount of fluid, and the very least leakage will materially affect the pressures. If, on the other hand, we have the piston so tight that it will not leak, we have so much friction that the results are very far from the truth. We have, in Sibley College, three dynamometers which were built to be worked on this plan, and we found the errors so very large that we have had to discard them, simply for the causes mentioned.

There is one other thing which he speaks of in the latter portion of the paper which, I think, would be very bad practice, and that is the use of mercury in contact with the indicator piston. I believe I have incidentally referred to the solvent action of mercury on brass in my paper on calibration of indicator springs. I think its use would certainly be exceedingly detrimental to the indicator, as in ordinary construction indicators are made of brass. It is true, steel indicators might be used, but these are not to be ordinarily had.

*Prof. W. S. Aldrich.**—The errors arising from using the indicator on a closed system have been pointed out in the body of the paper. The only curves so obtained were those of Fig. 35. These have not been integrated or otherwise used to obtain quantitative results. Such cards are, at best, as shown in Fig. 35, only of a comparative value, and that within a very small range of pressures.

The alternative method for use with a closed system has been suggested; namely, to use some form of diaphragm or closed tube pressure-recording gauge, instead of the indicator shown in Fig. 34. But, as the present paper dealt with dynamometric uses of only the indicator, the alternative (and better) method on a closed system, also employed by the author, has not been detailed. It would appear that the Sibley College dynamometers, which Professor Carpenter has spoken of as failing with the indicator on a closed system, might give excellent results by the use of diaphragm or closed tube pressure-recording gauges. As these are made for rectangular as well as circular diaphragms, the question of integrating the curves for the mean dynamometric pressures should not be a serious one.

Mercury inside the indicator is well recognized as bad practice, and the experiment is not here recorded as a precedent, but only as a bit of experience. As explained, before using the mercury the system comprising the "U" tube and indicator was filled with oil, after which the mercury was poured in. It was allowed to remain only a short while, and, as far as examined, worked no permanent injury to the indicator. Being the heaviest obtainable liquid, it was capable of holding the greatest pressure against the indicator piston before oozing past the same. Leakage sufficient to impair the records for quantitative measurements occurred, as reported, at about 28 lbs., in calibrating for the scale of pressures of Fig. 35.

* Author's closure, under the Rules.

DLXIV.*

THE CUMULATIVE ERRORS OF A GRADUATED SCALE.

BY W. A. ROGERS, WATERVILLE, ME.
(Member of the Society.)

WHEN the aliquot subdivisions of a graduated scale are compared *inter se*, each comparison may be given the same weight; but when the corrections determined are referred to the initial line by summing the series of relative corrections, the separate values of this summed series will evidently have different weights, since the effect of accidental errors of observation will increase with the number of summations. Thus, let $x_1, x_2, x_3, \ldots x_n$ represent the deviations of each of the spaces from the mean of all the spaces. These values of x will then represent the corrections required to reduce each space to the mean value of the n spaces without regard to the absolute value of the entire unit subdivided; in other words, they represent relative corrections. Referring these corrections to the graduations, x_1 will represent the correction required to reduce the second line to the position which it would have occupied if the subdivision of the entire unit into aliquot parts had been exact. The different values of x will therefore be entirely independent of each other and will each have the same weight for the same number of comparisons. It is evident that the separate values of x may be referred to the initial line of the scale by the process of summation. Thus x_1 represents the correction to be applied to line 2, line 1 being taken as the zero of reference; x_2 represents the correction to be applied to line 3, line 2 being taken as the zero of reference; hence, to connect the correction to line 3 with that for line 1 as the zero of reference, we have only to take the sum of x_1 and x_2. In the same manner we refer the corrections of the remaining graduations to the initial

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

line as a zero of reference by successive summations of the values which represent the relative corrections.

The method employed in measuring the relative errors of subdivisions of the scale may be described as the "stop method." The microscope is attached to a plate which is carried over the ways of the comparator by means of a rack and pinion movement. At each end of the plate is inserted a hardened steel plug, having a rounded end. Two movable sliding plates are placed upon the ways on opposite sides of the microscope-carriage. These plates are firmly clamped to the bed of the comparator, at such a distance apart that the carriage, in passing from contact with one stop to contact with the other stop, moves over a space approximately equal to one of the spaces to be compared. The microscope-carriage having been brought into contact with the left stop the micrometer line of the microscope is brought into coincidence with the initial line of the space taken as the unit of measurement. The carriage is then moved till contact is made with the right stop, and a reading of the microscope is taken for coincidence with the terminal line of the space. If the stops are set exactly equal to the space measured, the micrometer has no work to do. In practice, however, it is found advantageous to make the distance between the stops a little greater than the average scale distance.

The scale is then moved along upon the bed of the comparator parallel to itself, till a near coincidence with the first line of the second space is obtained, when the operation of measurement is repeated. In this way each successive space is compared with the constant distance between the stops. The method of procedure will appear from the following example:

	Spaces.	Left Stop. Div. of Microm.	Right Stop. Div. of Microm.	$(R - L)$	Relative Corrections.	Summed Series.
				div.	*div.*	*div.*
	1	4.4	8.8	+4.4	−0.4	−0.4
	2	6.0	9.9	+3.9	−0.9	−1.3
(1)	3	3.4	8.6	+5.2	+0.4	−0.9
	4	4.1	9.8	+5.7	+0.9	+0.0
	5	2.6	7.4	+4.8	+0.0	+0.0
				Mean +4.8		

Here + 4.8 divisions represents the excess of the distance between the stops over the average distance between the lines of the

graduated bar. Subtracting this mean from each value of $(R-L)$, we have the series of relative corrections required to reduce each line to the position which it would occupy if all the spaces were equal; but each space has a new line of reference.

In the last column, the values in the preceding column are all referred to the initial line of the scale. The summed series, therefore, indicated by the usual symbol Σ, represents the cumulative errors of the entire system of subdivisions compared. According to the conventional nomenclature adopted, a negative sign indicates that the measured space is too long.

Instead of the "stop method," one may employ with equal advantage the "two microscope method," when the spaces compared do not fall much below 6 or 8 centimeters. The "stop method" has, however, the advantage of allowing comparisons between spaces having any value whatever. With regard to the precision of which the method is capable, it may be said that the probable error of a single contact between the stops is decisively less than the probable error of a single measured coincidence with the filar micrometer.

The use of a screw in the measurement of cumulative errors requires that the periodic errors of the screw shall be definitely determined. Since errors of this class vary, not only with the load carried by the screw, but also with temperature and with the degree and kind of lubrication, this method is liable to lead to erroneous results, unless the greatest care is taken to secure the partial elimination of errors which it is possible to obtain in a limited degree by the method of comparison employed.

Judging by my own experience, the method is likely to give an appearance of precision which does not really exist.

In the application of the method here indicated, we are confronted with two sources of error, which baffle every effort at determination with perfect exactitude. They are the following:

(1) The values in the column of relative corrections, as shown in the example given, have the same weight, but the values in the summed series have different weights.

(2) The effect of an accidental error of observation at any point in the series of comparisons will be to give a wrong value to the mean, and this error will appear in the summed series in the form of a periodic error. Let us suppose that in the following fictitious example an accidental error, x, is introduced in the measurement of the fourth space. We shall then have:

Spaces.	$(R-L)$	Relative Corrections.	Summed Series.
1	+4 div.	$+0$ div. $-\frac{x}{10}$	$+0$ div. $-\frac{x}{10}$
2	+6	$+2 \quad -\frac{x}{10}$	$+2 \quad -\frac{2x}{10}$
3	+3	$-1 \quad -\frac{x}{10}$	$+1 \quad -\frac{3x}{10}$
4	$+4+x$	$+0 \quad +\frac{9x}{10}$	$+1 \quad +\frac{6x}{10}$
5	+2	$-2 \quad -\frac{x}{10}$	$-1 \quad +\frac{5x}{10}$
6	+1	$-3 \quad -\frac{x}{10}$	$-4 \quad +\frac{4x}{10}$
7	+3	$-1 \quad -\frac{x}{10}$	$-5 \quad +\frac{3x}{10}$
8	+7	$+3 \quad -\frac{x}{10}$	$-2 \quad +\frac{2x}{10}$
9	+2	$-2 \quad -\frac{x}{10}$	$-4 \quad +\frac{x}{10}$
10	+8	$+4 \quad -\frac{x}{10}$	$+0 \quad +0$
	Mean $= +4+\frac{x}{10}$		

It will be seen from this table that the form of the periodic error introduced in the summed series depends on the point at which the accidental error is introduced.

There is apparently another source of error, which has given the writer much trouble and which does not seem to admit of an easy explanation.

If the principal divisions are still further subdivided, the corrections derived for these subdivisions must be introduced into the principal summed series in such a manner that each correction shall be referred to the initial line. Let us suppose that in Example (1) each of the five divisions is still further subdivided into five parts, and that the following values of the subsidiary summed series have been obtained:

Spaces.	1	2	3	4	5
0	+0	+0	+0	+0	+0
1	+3	−4	−3	+2	+1
2	+2	+2	−4	+4	+3
3	+4	+3	−5	+4	+3
4	+3	+4	−1	+5	+4
5	+0	+0	+0	+0	+0

If we assume that the corrections to the principal subdivisions have been correctly determined, the effect of these errors upon the secondary subdivisions will be found by distributing the errors proportionately over the spaces covered.

The corrections for each line of the subsidiary series will then be found by adding the corresponding values in the two series. Thus, indicating the corrections to the principal subdivisions by brackets, we have:

Line.	Σ Principal Series.	Σ Secondary Series.	Σ Sum.	Line.	Σ Principal Series.	Σ Secondary Series.	Σ Sum.
0	+0.0 div.	+0.0 div.	+0.0 div.	13	−1.0 div.	−5.0 div.	−6.0 div.
1	−0.1	+3.0	+2.9	14	−1.0	−1.0	−2.0
2	−0.2	+2.0	+1.8	15	[−0.9]	+0.0	−0.9
3	−0.3	+4.0	+3.7	16	−0.7	+2.0	+1.3
4	−0.3	+3.0	+2.7	17	−0.5	+4.0	+3.5
5	[−0.4]	+0.0	−0.4	18	−0.4	+4.0	+3.6
6	−0.6	−4.0	−4.6	19	−0.2	+5.0	+4.8
7	−0.8	+2.0	+1.2	20	[+0.0]	+0.0	+0.0
8	−0.9	+3.0	+2.1	21	+0.0	+1.0	+1.0
9	−1.1	+4.0	+2.9	22	+0.0	+3.0	+3.0
10	[−1.3]	+0.0	−1.3	23	+0.0	+3.0	+3.0
11	−1.2	−3.0	−4.2	24	+0.0	+4.0	+4.0
12	−1.1	−4.0	−5.1	25	+0.0	+0.0	+0.0

If the scale has a correction for total length, this correction is to be distributed proportionately over the entire series.

The error to which attention is called appears in the transfer of graduations from one bar to another, when one applies in the process of ruling the corrections to the graduations of the primary bar, determined in the manner indicated above. For example, in the construction of a standard meter subdivided into 1,000 equal parts, it is always found that when the corrections to the primary subdivisions are large in value and irregular in sign, they appear in the transferred graduations in the form of errors of a periodic character reduced in magnitude. A similar result was found in the original division of a large index wheel into 1,000 equal parts. In both cases a third trial resulted in sets of graduations nearly free from errors of serious magnitude. In the case of the meter, a fourth trial gave the following results for the summed series.

CORRECTIONS.

	No. cases between 0.0μ and 1.0μ.	No. cases between 1.0μ and 2.0μ.	No. cases between 2.0μ and 3.0μ.	No. cases between 3.0μ and 4.0μ.
Decimeters	5	2	2	0
Centimeters	58	27	5	0
Millimeters	652	205	37	6

Reducing these values to ratios, and remembering that there are only 9 values in a summed series of 10 spaces, we have:

	No. cases between 0.0μ and 1.0μ.	No. cases between 1.0μ and 2.0μ.	No. cases between 2.0μ and 3.0μ.	No. cases between 3.0μ and 4.0μ.
Decimeters	0.44	0.33	0.23	0.00
Centimeters	0.64	0.31	0.05	0.00
Millimeters	0.72	0.23	0.04	0.01

A count of the number of relative errors in each group from which the summed series was derived, gave 856 cases in which the corrections fell between 0.0μ and 1.0μ.

It may be of interest to state that the time required for a complete investigation of the meter was about 80 hours.

It may be said that the existence of periodic errors, as shown in the transfer of graduations from one bar to another, may have been due to large accidental errors in the comparisons in the particular cases described, but the writer has obtained this result in a very large number of cases. In fact, no exception to this statement has so far been noted.

It may be mentioned in this connection that the principal difficulty in making an exact reproduction of a standard scale lies in the unequal depth of the cut of the ruling diamond at relatively hard and soft places in the metal. The writer has yet to find a metal whose difference of density at different points could not be detected in this way.

Admitting, then, the existence of at least two of the three classes of errors described, it is evident that their magnitude must be determined before an estimate can be made of the real precision of the comparisons.

Perhaps no more conclusive evidence could be obtained than by comparing the results obtained for a given scale in the man-

ner here indicated, with those obtained for the same standard by the method of measurement and investigation given by Dr. Broch in Volume V. of the "Memoirs of the International Bureau of Weights and Measures." The standard selected for comparison is a decimeter upon speculum metal, which is made the basis of all transfers to other bars of the same metal. It is designated $S_{\frac{m}{10}}$.

Inasmuch as the theoretical discussion given on pages 5 to 17 of the "Memoirs" is not readily available, the following *résumé* is given:

Let $x_1, x_2, x_3, \ldots x_{n+1}$ be the corrections required to be applied to the successive centimeter graduations of the decimeter scale.

Let $\lambda_n, \lambda_{n-1}, \lambda_{n-2}$, etc., be the corrections to an assumed unit with which each of the successive spaces is compared.

Let $a_1, a_2, a_3, \ldots a_n$; $b_1, b_2, b_3, \ldots b_{n-1}$; $c_1, c_2, c_3, \ldots c_{n-2}$, etc., represent the numerical values of the corrections obtained by comparing each of the successive spaces with the units selected. We then have (see page 5):

(1)

$$\begin{aligned}
\lambda_n + x_1 - x_2 &= a_1 \\
\lambda_n + x_2 - x_3 &= a_2 \\
\lambda_n + x_3 - x_4 &= a_3 \\
&\cdots\cdots \\
\lambda_n + x_n - x_{n+1} &= a_n
\end{aligned}$$

(2)

$$\begin{aligned}
\lambda_{n-1} + x_1 - x_3 &= b_1 \\
\lambda_{n-1} + x_2 - x_4 &= b_2 \\
\lambda_{n-1} + x_3 - x_5 &= b_3 \\
&\cdots\cdots \\
\lambda_{n-1} + x_{n-1} - x_{n+1} &= b_{n-1}
\end{aligned}$$

(3)

$$\begin{aligned}
\lambda_{n-2} + x_1 - x_4 &= c_1 \\
\lambda_{n-2} + x_2 - x_5 &= c_2 \\
\lambda_{n-2} + x_3 - x_6 &= c_3 \\
&\cdots\cdots \\
\lambda_{n-2} + x_{n-2} - x_{n+1} &= c_{n-2}
\end{aligned}$$

(4)

$$\begin{aligned}
\lambda_{n-3} + x_1 - x_5 &= d_1 \\
\lambda_{n-3} + x_2 - x_6 &= d_2 \\
\lambda_{n-3} + x_3 - x_7 &= d_3 \\
&\cdots\cdots \\
\lambda_{n-3} + x_{n-3} - x_{n+1} &= d_{n-3}
\end{aligned}$$

(5)

$$\begin{aligned}
\lambda_{n-4} + x_1 - x_6 &= e_1 \\
\lambda_{n-4} + x_2 - x_7 &= e_2 \\
\lambda_{n-4} + x_3 - x_8 &= e_3 \\
&\cdots\cdots \\
\lambda_{n-4} + x_{n-4} - x_{n+1} &= e_{n-4}
\end{aligned}$$

(6)

$$\begin{aligned}
\lambda_{n-5} + x_1 - x_7 &= f_1 \\
\lambda_{n-5} + x_2 - x_8 &= f_2 \\
\lambda_{n-5} + x_3 - x_9 &= f_3 \\
&\cdots\cdots \\
\lambda_{n-5} + x_{n-5} - x_{n+1} &= f_{n-5}
\end{aligned}$$

(7)

$$\lambda_{n-6} + x_1 - x_8 = g_1$$
$$\lambda_{n-6} + x_2 - x_9 = g_2$$
$$\lambda_{n-6} + x_3 - x_{10} = g_3$$
$$\cdots\cdots\cdots$$
$$\lambda_{n-6} + x_{n-6} - x_{n+1} = g_{n-6}$$

(8)

$$\lambda_{n-7} + x_1 - x_9 = h_1$$
$$\lambda_{n-7} + x_2 - x_{10} = h_2$$
$$\lambda_{n-7} + x_3 - x_{11} = h_3$$

(9)

$$\lambda_{n-8} + x_1 - x_{10} = i_1$$
$$\lambda_{n-8} + x_2 - x_{11} = i_2$$

These equations are very conveniently arranged in Table I., on page 8. The numerical results of the comparisons of the subdivisions of our scale will be subsequently given in this form.

It will be seen that the coefficients of the unknown quantities are in every case unity; hence the solution of the equations by least squares is simply a process of summation resulting in the following normal equations:

$$S_n = n\lambda_n + x_1 - x_{n+1};$$

$$S_{n-1} = (n-1)\,\lambda_{n-1} + (x_1 + x_2) - (x_n + x_{n+1});$$

$$S_{n-2} = (n-2)\,\lambda_{n-2} + (x_1 + x_2 + x_3) - (x_{n-1} + x_n + x_{n+1});$$

$$\cdots\cdots\cdots\cdots\cdots\cdots$$

$$S_{\frac{n}{2}+1} = \left(\frac{n}{2} + 1\right)\lambda_{\frac{n}{2}+1} + (x_1 + x_2 + x_3 + \ldots + x_{\frac{n}{2}})$$
$$- (x_{\frac{n}{2}+2} + x_{\frac{n}{2}+3} + \ldots\ldots\ldots + x_{n+1});$$

$$S_{\frac{n}{2}} = \frac{n}{2}\lambda_{\frac{n}{2}} + (x_1 + x_2 + x_3 + \ldots + x_{\frac{n}{2}})$$
$$- (x_{\frac{n}{2}+2} + x_{\frac{n}{2}+3} + \ldots\ldots\ldots + x_{n+1});$$

$$\cdots\cdots\cdots\cdots\cdots\cdots$$

$$S_3 = 3\lambda_3 + (x_1 + x_2 + x_3);$$

$S_2 = 2\lambda_2 + (x_1 + x_2)$;
$0 = \lambda_1 + x_1$.

Table II. on page 9 contains the summations $a_1 + b_1 + c_1 +$ etc. $= t_1$; $a_2 + b_2 + c_2 +$ etc. $= t_2$; $a_3 + b_3 + c_3 +$ etc. $= t_3$, etc., etc. As will be seen from the numerical example to be given, the values of $t_1, t_2, t_3, \ldots t_{n+1}$ are found by writing in vertical columns the values in the horizontal columns in Table I., and also, with the opposite sign, those values of x which have the same subscript for $-x$ as that given at the top of the column in Table II. A short line is drawn at the point of demarkation.

The solution of the normal equations is facilitated by the elegant transformations given on pages 9 to 16. The following are the resulting equations employed in the solution:

$$X = x_1 + x_2 + x_3 + \ldots + x_{n-1} + x_n + x_{n+1} \quad . \quad . \quad (1)$$

Control for the values of t and for the sums S given in Table I.:

$$t_1 + t_2 + t_3 + \ldots + t_{n-2} + t_n + t_{n+1} = 0.$$

$$nt_1 + (n-2)\,t_2 + (n-4)\,t_3 + \ldots - (n-4)\,t_{n-1}$$
$$- (n-2)\,t_n - nt_{n+1} = 2[Sn + 2S_{n-1} + 3S_{n-2} + \ldots$$
$$+ (n-2)\,S_3 + (n-1)\,S_2].$$

Relations between S, P, and X, given by equations (9) on page 12, and equations (6) on page 11:

$$\begin{array}{ll} S_{11} = t_{11} + t_1 & 11P_{11} = S_{11} + 2X \\ S_{10} = t_{10} + t_2 & 11P_{10} = S_{10} + 2X \\ S_9 = t_9 + t_3 & 11P_9 = S_9 + 2X \\ S_8 = t_8 + t_4 & 11P_8 = S_8 + 2X \\ S_7 = t_7 + t_5 & 11P_7 = S_7 + 2X \\ S_6 = t_6 + t_6 & 11P_6 = S_6 + 2X \end{array} \qquad 2X = 11P_{11} - S_{11}$$

Control:

$$P_{11} + P_{10} + P_9 + P_8 + P_7 + \frac{P_6}{2} = X.$$

Relations between D, R, and Σ, from equations (10) on page 13, and equations (16) on page 14:

$$\begin{array}{ll} D_{11} = t_{11} - t_1 & 11R_{10} = D_{10} - D_{11} - \Sigma_{10} \\ D_{10} = t_{10} - t_2 & 11R_9 = D_9 - D_{10} - \Sigma_9 \\ D_9 = t_9 - t_3 & 11R_8 = D_8 - D_9 - \Sigma_8 \\ D_8 = t_8 - t_4 & 11R_7 = D_7 - D_8 - \Sigma_7 \\ D_7 = t_7 - t_5 & \end{array}$$

In which the values of Σ are derived from the following equations, given on page 12:

$$\Sigma_n = 2\,\frac{S_n}{n},$$

$$\Sigma_{n-1} = 2\left(\frac{S_{n-1}}{n-1} + \frac{S_2}{2}\right),$$

$$\Sigma_{n-2} = 2\left(\frac{S_{n-2}}{n-2} + \frac{S_3}{3}\right),$$

$$\Sigma_{\frac{n+2}{2}} = 2\left(\frac{S_{\frac{n}{2}+2}}{\frac{n}{2}+2} + \frac{S_{\frac{n}{2}-1}}{\frac{n}{2}-1}\right).$$

Relations between Q and R from equations (17) on page 15:

$$\begin{aligned}
Q_{10} &= R_{10} + Q_{11} - \tfrac{1}{5}\ Q_{11}\\
Q_9 &= R_9 + Q_{10} - \tfrac{1}{9}\,(Q_{11} + Q_{10})\\
Q_8 &= R_8 + Q_9 - \tfrac{1}{12}\,(Q_{11} + Q_{10} + Q_9)\\
Q_7 &= R_7 + Q_8 - \tfrac{1}{14}\,(Q_{11} + Q_{10} + Q_9 + Q_8)
\end{aligned}$$

Relations between P, Q, and the values of x from equations (18) on page 15:

$$\begin{aligned}
x_{11} &= \tfrac{1}{2}\,(P_{11} + Q_{11}) & x_5 &= \tfrac{1}{2}\,(P_7 - Q_7)\\
x_{10} &= \tfrac{1}{2}\,(P_{10} + Q_{10}) & x_4 &= \tfrac{1}{2}\,(P_8 - Q_8)\\
x_9 &= \tfrac{1}{2}\,(P_9 + Q_9) & x_3 &= \tfrac{1}{2}\,(P_9 - Q_9)\\
x_8 &= \tfrac{1}{2}\,(P_8 + Q_8) & x_2 &= \tfrac{1}{2}\,(P_{10} - Q_{10})\\
x_7 &= \tfrac{1}{2}\,(P_7 + Q_7) & x_1 &= \tfrac{1}{2}\,(P_{11} - Q_{11})\\
x_6 &= \tfrac{1}{2}\,(P_6 + Q_6)
\end{aligned}$$

Control:

$$\Sigma x = \tfrac{1}{2}\,(11P_{11} - S_{11}).$$

Relations between λ and Q, from equations (20) on page 15:

$$\begin{aligned}
\lambda_2 &= \tfrac{1}{2}\,(Q_{11} + Q_{10} + S_2)\\
\lambda_3 &= \tfrac{1}{3}\,(Q_{11} + Q_{10} + Q_9 + S_3)\\
\lambda_4 &= \tfrac{1}{4}\,(Q_{11} + Q_{10} + Q_9 + Q_8 + S_4)\\
\lambda_5 &= \tfrac{1}{5}\,(Q_{11} + Q_{10} + Q_9 + Q_8 + Q_7 + S_5)\\
\lambda_6 &= \tfrac{1}{6}\,(Q_{11} + Q_{10} + Q_9 + Q_8 + Q_7 + S_6)\\
\lambda_7 &= \tfrac{1}{7}\,(Q_{11} + Q_{10} + Q_9 + Q_8 + S_7)\\
\lambda_8 &= \tfrac{1}{8}\,(Q_{11} + Q_{10} + Q_9 + S_8)\\
\lambda_9 &= \tfrac{1}{9}\,(Q_{11} + Q_{10} + S_9)\\
\lambda_{10} &= \tfrac{1}{10}\,(Q_{11} + S_{10})
\end{aligned}$$

Control:

$$D_7 = t_7 - t_5 = 11Q_7 - 2\,(\lambda_5 + \lambda_6).$$

In obtaining the following data, the stops of the comparator were set nearly equal to the average distance between the centimeter graduations. In the preliminary solution of the equations, the constant obtained by taking the mean of the values $(R - L)$ of the micrometer readings was subtracted from the separate values of the series, thus avoiding the use of the quantities designated by S_{10}, S_9, S_8, etc. In order to follow strictly the form of computation in the "Memoirs," the original values of $(R - L)$ are here retained.

DATA FROM OBSERVATION.

$x_1 - x_2 = -0.5\mu$	$x_1 - x_3 = +0.0\mu$	$x_1 - x_4 = -0.6\mu$	$x_1 - x_5 = +0.2\mu$	$x_1 - x_6 = +1.2\mu$	$x_1 - x_7 = -0.2\mu$	$x_1 - x_8 = +0.7\mu$	$x_1 - x_9 = -0.7\mu$	$x_1 - x_{10} = -0.9\mu$
$x_2 - x_3 = +0.0$	$x_2 - x_4 = +0.2$	$x_2 - x_5 = +0.6$	$x_2 - x_6 = +2.5$	$x_2 - x_7 = -0.2$	$x_2 - x_8 = +0.9$	$x_2 - x_9 = +0.3$	$x_2 - x_{10} = -0.2$	$x_2 - x_{11} = +0.1$
$x_3 - x_4 = +0.0$	$x_3 - x_5 = +0.6$	$x_3 - x_6 = +2.1$	$x_3 - x_7 = +0.9$	$x_3 - x_8 = +0.7$	$x_3 - x_9 = -0.3$	$x_3 - x_{10} = -0.2$	$x_3 - x_{11} = +0.3$	
$x_4 - x_5 = +0.2$	$x_4 - x_6 = +2.4$	$x_4 - x_7 = +0.7$	$x_4 - x_8 = +1.3$	$x_4 - x_9 = -0.6$	$x_4 - x_{10} = -0.5$	$x_4 - x_{11} = +0.0$		
$x_5 - x_6 = +1.5$	$x_5 - x_7 = +0.6$	$x_5 - x_8 = +1.0$	$x_5 - x_9 = -0.3$	$x_5 - x_{10} = -0.8$	$x_5 - x_{11} = -0.4$			
$x_6 - x_7 = -1.5$	$x_6 - x_8 = -0.9$	$x_6 - x_9 = -2.0$	$x_6 - x_{10} = -2.0$	$x_6 - x_{11} = -2.7$				
$x_7 - x_8 = +0.2$	$x_7 - x_9 = -0.4$	$x_7 - x_{10} = -0.7$	$x_7 - x_{11} = +0.2$					
$x_8 - x_9 = -1.1$	$x_8 - x_{10} = -1.2$	$x_8 - x_{11} = -0.3$						
$x_9 - x_{10} = -0.2$	$x_9 - x_{11} = +0.5$							
$x_{10} - x_{11} = +0.4$								
$S_{10} = -1.00$	$S_9 = +1.80$	$S_8 = +0.80$	$S_7 = +2.80$	$S_6 = -2.40$	$S_5 = -0.50$	$S_4 = +0.80$	$S_3 = -0.60$	$S_2 = -0.80$
$\frac{2S_{10}}{10} = -0.20$	$\frac{2S_9}{9} = +0.40$	$\frac{2S_8}{8} = +0.20$	$\frac{2S_7}{7} = +0.80$	$\frac{2S_6}{6} = -0.80$	$\frac{2S_5}{5} = -0.20$	$\frac{2S_4}{4} = +0.40$	$\frac{2S_3}{3} = -0.40$	$\frac{2S_2}{2} = -0.80$
$\Sigma_{10} = -0.20$	$\Sigma_9 = -0.40$	$\Sigma_8 = -0.20$	$\Sigma_7 = +1.20$					

COMPUTATION OF $t_1, t_2, t_3, \ldots t_{11}$.

	x_1	x_2	x_3	x_4	x_5	x_6	x_7	x_8	x_9	x_{10}	x_{11}
	−0.5 μ	+0.0 μ	+0.0 μ	+0.2 μ	+1.5 μ	− 1.5 μ	+0.2 μ	−1.1 μ	−0.2 μ	+0.4 μ	+0.0 μ
	+0.0	+0.2	+0.6	+2.4	+0.6	− 0.9	−0.4	−1.2	+0.	+0.9	−0.1
	−0.6	+0.6	+2.1	+0.7	+1.0	− 2.0	−0.7	−0.3	+0.7	+0.2	−0.3
	+0.2	+2.5	+0.9	+1.8	−0.3	− 2.0	+0.2	−0.7	−0.3	+0.5	+0.0
	+1.2	−0.2	+0.7	−0.6	−0.8	− 2.7	+0.2	−0.9	+0.3	+0.8	+0.4
	−0.2	+0.9	−0.3	−0.5	−0.4	− 1.2	+0.2	−0.7	+0.6	+2.0	+2.7
	+0.7	+0.3	−0.2	+0.0	−0.2	− 2.5	−0.9	−1.3	+0.3	+0.7	−0.2
	−0.7	−0.2	+0.3	+0.6	−0.6	− 2.1	−0.7	−1.0	+2.0	+1.2	+0.3
	−0.9	+0.1	+0.0	−0.2	−0.6	− 2.4	−0.6	+0.9	+0.4	+0.2	−0.5
		+0.5	+0.0	+0.0	−0.2	− 1.5	+1.5	−0.2	+1.1	+0.2	−0.4
Sums:	−0.8	+4.7	+4.1	+3.9	+0.0	−18.8	−1.0	−6.5	+5.4	+7.1	+1.9
	t_1	t_2	t_3	t_4	t_5	t_6	t_7	t_8	t_9	t_{10}	t_{11}

Control: $\Sigma t = 0$.

Control:

$$
\begin{aligned}
5t_1 &= & -4.0 \quad - t_7 &= +1.0 \\
4t_2 &= +18.8 & -2t_8 &= +13.0 \\
3t_3 &= +12.3 & -3t_9 &= -16.2 \\
2t_4 &= +7.8 & -4t_{10} &= -28.4 \\
t_5 &= +0.0 & -5t_{11} &= -9.5 \\
& +34.9 & -40.1 & \\
& -40.1 & & \\
\textbf{Sum:} & -5.2 & &
\end{aligned}
$$

$$
\begin{aligned}
9S_2 &= & -7.2 \quad 4S_7 &= +11.2 \\
8S_3 &= & -4.8 \quad 3S_8 &= +2.4 \\
7S_4 &= +5.6 & 2S_9 &= +3.6 \\
6S_5 &= & -3.0 & \\
5S_6 &= & -12.0 \quad S_{10} &= -1.0 \\
\textbf{Sum:} & & -5.2 &
\end{aligned}
$$

COMPUTATION OF P.

$t_{11} = +1.9$	$t_{10} = +7.1$	$t_9 = +5.4$
$t_1 = -0.8$	$t_2 = +4.7$	$t_3 = +4.1$
$S_{11} = +1.1$	$S_{10} = +11.8$	$S_9 = +9.5$
$2X = -1.1$	$2X = -1.1$	$2X = -1.1$
$11P_{11} = +0.0$	$11P_{10} = +10.7$	$11P_9 = +8.4$
$P_{11} = +0.0$	$P_{10} = +0.97$	$P_9 = +0.76$

$t_8 = -6.5$	$t_7 = -1.0$	$t_6 = -18.8$
$t_4 = +3.9$	$t_5 = +0.0$	$t_6 = -18.8$
$S_8 = -2.6$	$S_7 = -1.0$	$S_6 = -37.6$
$2X = -1.1$	$2X = -1.1$	$2X = -1.1$
$11P_8 = -3.7$	$11P_7 = -2.1$	$11P_6 = -38.7$
$P_8 = -0.34$	$P_7 = -0.19$	$P_6 = -3.52$

Control:

$$P_{11} + P_{10} + P_9 + P_8 + P_7 + \frac{P_6}{2} = -0.56$$

$$X = -0.55$$

COMPUTATION OF D AND R.

$t_{11} = +1.9$	$t_{10} = +7.1$	$t_9 = +5.4$
$-t_1 = +0.8$	$-t_2 = -4.7$	$-t_3 = -4.1$
$D_{11} = +2.7$	$D_{10} = +2.4$	$D_9 = +1.3$
	$-D_{11} = -2.7$	$-D_{10} = -2.4$
	$-\Sigma_{10} = +0.2$	$-\Sigma_9 = +0.4$
	$11R_{10} = -0.1$	$11R_9 = -0.7$
	$R_{10} = -0.01$	$R_9 = -0.06$

$t_8 = -6.5$	$t_7 = -1.0$
$-t_4 = -3.9$	$-t_5 = +0.0$
$D_8 = -10.4$	$D_7 = -1.0$
$-D_9 = -1.3$	$-D_8 = +10.4$
$-\Sigma_8 = +0.2$	$-\Sigma_7 = -1.2$
$11R_8 = -11.5$	$11R_7 = +8.2$
$R_8 = -1.04$	$R_7 = +0.74$

COMPUTATION OF Q.

$R_{10} = -0.01$	$R_9 = -0.06$
$Q_{11} = 0.00$	$Q_{10} = -0.01$
$-\frac{1}{6} Q_{11} = 0.00$	$-\frac{1}{9} \Sigma Q = +0.00$
$Q_{10} = -0.01$	$Q_9 = -0.07$

$R_8 = -1.04$	$R_7 = +0.74$
$Q_9 = -0.07$	$Q_8 = -1.10$
$-\frac{1}{12} \Sigma Q = +0.01$	$-\frac{1}{14} \Sigma Q = +0.08$
$Q_8 = -1.10$	$Q_7 = -0.28$

COMPUTATION OF $x_1, x_2, x_3, \ldots x_{11}$.

Index.	P	Q	$P+Q$	$P-Q$
11	+ 0.00	+ 0.00	+ 0.00	+ 0.00
10	+ 0.97	− 0.01	+ 0.96	+ 0.98
9	+ 0.76	− 0.07	+ 0.69	+ 0.83
8	− 0.34	− 1.10	− 1.44	+ 0.76
7	− 0.19	− 0.28	− 0.47	+ 0.09
6	− 3.52	. . . .	− 3.52	

$x_{11} = +0.00$ $\quad$ $x_1 = +0.00$

$x_{10} = +0.48$ $\quad$ $x_2 = +0.49$

$x_9 = +0.34$ $\quad$ $x_3 = +0.41$

$x_8 = -0.72$ $\quad$ $x_4 = +0.38$

$x_7 = -0.23$ $\quad$ $x_5 = +0.04$

$x_6 = -1.76$

Control:

$\Sigma x = -0.57.$ $\qquad$ $\frac{1}{2}11P_{11} - S_{11} = -0.55.$

COMPUTATION OF $\lambda_2, \lambda_3, \lambda_4, \ldots \lambda_{10}$.

$Q_{11} = +0.00$ $\quad$ $\Sigma_{10}Q = -0.01$ $\quad$ $\Sigma_9 Q = -0.08$ $\quad$ $\Sigma_8 Q = -1.18$

$Q_{10} = -0.01$ $\quad$ $Q_9 = -0.07$ $\quad$ $Q_8 = -1.10$ $\quad$ $Q_7 = -0.28$

$\Sigma_{10} = -0.01$ $\quad$ $\Sigma_9 Q = -0.08$ $\quad$ $\Sigma_8 Q = -1.18$ $\quad$ $\Sigma_7 Q = -1.46$

$S_2 = -0.80$ $\quad$ $S_3 = -0.60$ $\quad$ $S_4 = +0.80$

$\Sigma_{10}Q = -0.01$ $\quad$ $\Sigma_9 Q = -0.08$ $\quad$ $\Sigma_8 Q = -1.18$

$2\lambda_2 = -0.81$ $\quad$ $3\lambda_3 = -0.68$ $\quad$ $4\lambda_4 = -0.38$

$\lambda_2 = -0.40$ $\quad$ $\lambda_3 = -0.23$ $\quad$ $\lambda_4 = -0.09$

$S_5 = -0.50$ $\quad$ $S_6 = -2.40$ $\quad$ $S_7 = +2.80$

$\Sigma_{10}Q = -1.46$ $\quad$ $\Sigma_7 Q = -1.46$ $\quad$ $\Sigma_8 Q = -1.18$

$5\lambda_5 = -1.96$ $\quad$ $6\lambda_6 = -3.88$ $\quad$ $7\lambda_7 = +1.62$

$\lambda_5 = -0.39$ $\quad$ $\lambda_6 = -0.64$ $\quad$ $\lambda_7 = +0.23$

$S_8 = +0.80$ $\quad$ $S_9 = +1.80$ $\quad$ $S_{10} = -1.00$

$8\lambda_8 = +0.72$ $\quad$ $9\lambda_9 = +1.79$ $\quad$ $10\lambda_{10} = -1.00$

$\lambda_8 = +0.09$ $\quad$ $\lambda_9 = +0.20$ $\quad$ $\lambda_{10} = -0.10$

COMPARISON OF CALCULATION WITH OBSERVATION.

$x_1=0.0$, $x_2=+0.49$; $x_3=+0.41$; $x_4=+0.38$; $x_5=+0.04$; $x_6=-1.76$; $x_7=-0.23$; $x_8=-0.72$; $x_9=+0.34$; $x_{10}=+0.48$; $x_{11}=0.00$.

$\lambda_2=-0.40$; $\lambda_3=-0.23$; $\lambda_4=-0.09$; $\lambda_5=-0.39$; $\lambda_6=-0.64$; $\lambda_7=+0.23$; $\lambda_8=+0.09$; $\lambda_9=+0.20$; $\lambda_{10}=-0.10$.

λ_{10}	*compt.*	*obs.*	*c−o*
$x_1-x_2=$	−0.49	−0.10	−0.59−0.50+0.09 μ
$x_2-x_3=$	+0.08	−.10	−0.02+0.00−0.02
$x_3-x_4=$	+0.03	−.10	−0.07+0.00−0.07
$x_4-x_5=$	+0.34	−.10	+0.24+0.20+0.04
$x_5-x_6=$	+1.80	−.10	+1.70+1.50+0.20
$x_6-x_7=$	−1.53	−.10	−1.63−1.50−0.13
$x_7-x_8=$	+0.49	−.10	+0.39+0.20+0.19
$x_8-x_9=$	−1.06	−.10	−1.16−1.10−0.06
$x_9-x_{10}=$	−0.14	−.10	−0.24−0.20−0.04
$x_{10}-x_{11}=$	+0.48	−.10	+0.38+0.40−0.02

λ_9	*compt.*	*obs.*	*c−o*
$x_1-x_3=$	−0.41	+0.20	−0.21+0.00−0.21 μ
$x_2-x_4=$	+0.11	+.20	+0.31+0.20+0.11
$x_3-x_5=$	+0.37	+.20	+0.57+0.60−0.03
$x_4-x_6=$	+2.14	+.20	+2.34+2.40−0.06
$x_5-x_7=$	+0.27	+.20	+0.47+0.60−0.13
$x_6-x_8=$	−1.04	+.20	−0.84−0.90+0.06
$x_7-x_9=$	−0.57	+.20	−0.37−0.40+0.03
$x_8-x_{10}=$	−1.20	+.20	−1.00−0.20+1.20
$x_9-x_{11}=$	+0.48	+.20	+0.68+1.50+0.18

λ_8			
$x_1-x_4=$	−0.38	+0.09	−0.28−0.60−0.32 μ
$x_2-x_5=$	+0.45	+.09	+0.54+0.60−0.46
$x_3-x_6=$	+2.17	+.09	+2.26+2.10+0.16
$x_4-x_7=$	+0.61	+.09	+0.70+0.70+0.00
$x_5-x_8=$	+0.76	+.09	+0.85+1.00−0.15
$x_6-x_9=$	−2.10	+.09	−2.01−2.00−0.01
$x_7-x_{10}=$	−0.71	+.09	−0.62−0.70+0.08
$x_8-x_{11}=$	−0.72	+.09	−0.63−0.30−0.33

λ_7			
$x_1-x_5=$	−0.04	+0.23	+0.19+0.20−0.01 μ
$x_2-x_6=$	+2.25	+.23	+2.48+2.50−0.02
$x_3-x_7=$	+0.64	+.23	+0.87+0.90−0.03
$x_4-x_8=$	+1.10	+.23	+1.33+1.30+0.03
$x_5-x_9=$	−0.30	+.23	−0.07−0.30+0.28
$x_6-x_{10}=$	−2.24	+.23	−2.01−2.00−0.01
$x_7-x_{11}=$	−0.23	+.23	+0.00+0.20−0.20

λ_6			
$x_1-x_6=$	+1.96	−0.64	+1.12+1.20−0.08 μ
$x_2-x_7=$	+0.72	−.64	+0.08−0.20+0.28
$x_3-x_8=$	+1.13	−.64	+0.49+0.70−0.21
$x_4-x_9=$	+0.04	−.64	−0.60−0.60+0.00
$x_5-x_{10}=$	−0.44	−.64	−1.08−0.80−0.28
$x_6-x_{11}=$	−1.76	−.64	−2.40−2.70+0.30

λ_5			
$x_1-x_7=$	+0.23	−0.39	−0.16−0.20+0.04 μ
$x_2-x_8=$	+1.21	−.39	+0.82+0.90−0.08
$x_3-x_9=$	+0.07	−.39	−0.32−0.30−0.02
$x_4-x_{10}=$	−0.10	−.39	−0.49−0.50+0.01
$x_5-x_{11}=$	+0.04	−.39	−0.35−0.40+0.05

λ_4			
$x_1-x_8=$	+0.72	−0.09	+0.63+0.70−0.07 μ
$x_2-x_9=$	+0.15	−.09	+0.06+0.30−0.24
$x_3-x_{10}=$	−0.07	−.09	−0.16−0.20+0.04
$x_4-x_{11}=$	+0.38	−.09	+0.29+0.00+0.29

λ_3			
$x_1-x_9=$	−0.34	−0.23	−0.57−0.70+0.13 μ
$x_2-x_{10}=$	+0.01	−.23	−0.22−0.20−0.02
$x_3-x_{11}=$	+0.41	−.23	+0.18+0.30−0.12

λ_2	*compt.*	*obs.*	*c−o*
$x_1-x_{10}=$	−0.48	−0.40	−0.88−0.90+0.02 μ
$x_2-x_{11}=$	+0.49	−.40	+0.09+0.10−0.01

The value of the probable error of a single space-determination derived from all the residuals is $r_0 = \pm\, 0.092\mu$. Even admitting that there is no elimination of accidental errors in the summation for five spaces, it appears that the middle value will have no error much exceeding 0.5μ. This result is in close agreement with my ordinary experience. When the spaces compared are decimeters, the value of r_0 will be slightly increased, but it rarely reaches 0.15μ.

We are now prepared to make a direct comparison of the values of x with corresponding values derived from the summation of the values in the column $x_1 - x_2$, $x_2 - x_3$, etc. In order to make this comparison under the nomenclature adopted in the latter form of reduction, the signs of the values of x derived above are to be changed.

Line.	$R-L$	Relative Corrections.	Σ	From discussion.	Δ
2	$-0.5\ \mu$	$-0.4\ \mu$	$-0.40\ \mu$	$-0.49\ \mu$	$+0.09$
3	$+0.0$	$+0.1$	-0.30	-0.41	$+0.11$
4	$+0.0$	$+0.1$	-0.20	-0.38	$+0.18$
5	$+0.2$	$+0.3$	$+0.10$	-0.04	$+0.14$
6	$+1.5$	$+1.6$	$+1.70$	$+1.76$	-0.06
7	-1.5	-1.4	$+0.30$	$+0.23$	$+0.07$
8	$+0.2$	$+0.3$	$+0.60$	$+0.72$	-0.12
9	-1.1	-1.0	-0.40	-0.34	-0.06
10	-0.2	-0.1	-0.50	-0.48	-0.02
11	$+0.4$	$+0.5$	$+0.00$	$+0.00$	$+0.00$
	Mean -0.10				

It appears, therefore, that the shorter method gives results which differ but little from those obtained from the more elaborate discussion, the average difference being only 0.09 μ. It will be observed, however, that the residuals have a periodic form; a result which may always be expected in any summed series. I add here the results obtained by two students in the college, Mr. D. W. Kimball and Mr. F. E. Russell.

Line.	Kimball. Σ	Russell. Σ	Mean.
1	$-0.7\ \mu$	$-0.5\ \mu$	$-0.60\ \mu$
2	-0.4	-0.3	-0.35
3	-0.4	-0.1	-0.25
4	-0.4	$+0.1$	-0.15
5	$+1.9$	$+1.8$	$+1.85$
6	$+0.8$	$+0.7$	$+0.75$
7	$+0.4$	$+0.6$	$+0.50$
8	-0.6	-0.4	-0.50
9	-0.4	-0.6	-0.50
10	$+0.0$	$+0.0$	$+0.00$

The results obtained from a comparison of the aliquot subdivisions of a given unit may be verified in the following manner:

In subdivisions of metric units we may compare the two halves of the entire unit—*e.g.*, of 1 decimeter—and then combine the results with those obtained by comparing the separate subdivisions of each half.

If English units are employed, several combinations can be formed.

As an illustration, I give here the results of the discussion of a combined decimeter and four-inch standard upon speculum metal,

constructed for Mr. J. A. Brashear upon an order from Dr. Thalen, of Upsala.

The decimeters are subdivided to millimeters, and the last millimeter is further subdivided to $\frac{1}{10}$ *mm.* and $\frac{1}{100}$ *mm.* The four-inch scale is subdivided to tenths of inches. The last inch is subdivided into 100 equal parts, and the last hundredth is subdivided to $\frac{1}{1000}$ inch.

CORRECTIONS TO THE MILLIMETER SUBDIVISIONS.

In this discussion only two combinations are made, viz.: (1) Centimeters : millimeters of each centimeter. (2) ½ decimeter : centimeters of each ½ decimeter ; millimeters of each centimeter.

The means of the two results are given in column (*a*), and the deviations of the mean from either system are given in column (*b*). The corrections are referred to the initial line.

	(*a*)	(*b*)		(*a*)	(*b*)		(*a*)	(*b*)		(*a*)	(*b*)
0.	+0.0 μ	+0.0 μ	14.	+1.1 μ	+0.3 μ	28.	+0.9 μ	+0.0 μ	42.	+0.9 μ	−0.1 μ
1.	+1.3	+0.1	15.	+0.8	+0.3	29.	+1.3	+0.3	43.	+1.0	−0.1
2.	+1.4	+0.0	16.	+1.4	+0.2	30.	+0.5	+0.1	44.	+0.6	−0.2
3.	+1.6	+0.2	17.	+1.1	+0.3	31.	+1.1	+0.1	45.	+1.1	−0.2
4.	+1.2	+0.3	18.	+1.7	+0.6	32.	+0.9	+0.1	46.	+1.5	−0.2
5.	+1.6	+0.2	19.	+1.3	+0.0	33.	+1.2	+0.0	47.	+0.8	−0.2
6.	+1.6	+0.3	20.	+1.1	+0.1	34.	+1.7	+0.1	48.	+0.2	−0.2
7.	+1.5	+0.4	21.	+1.6	+0.0	35.	+0.5	+0.1	49.	+0.8	−0.2
8.	+1.6	+0.4	22.	+1.6	−0.3	36.	+0.9	−0.1	50.	+0.8	−0.1
9.	+0.5	+0.3	23.	+1.6	+0.0	37.	+0.7	+0.1	51.	+0.3	−0.3
10.	+0.2	+0.3	24.	+1.6	−0.4	38.	+1.1	+0.1	52.	+0.6	+0.0
11.	+1.1	+0.3	25.	+1.4	−0.3	39.	+1.1	+0.1	53.	+1.0	+0.0
12.	+1.7	+0.3	26.	+1.1	−0.3	40.	+0.6	−0.1	54.	+1.6	+0.2
13.	+1.1	+0.3	27.	+1.1	−0.3	41.	+2.2	+0.0	55.	+1.5	−0.1

	(*a*)	(*b*)		(*a*)	(*b*)		(*a*)	(*b*)		(*a*)	(*b*)
56.	+1.1 μ	+0.0 μ	67.	−0.2 μ	+0.2 μ	78.	−0.4 μ	+0.3 μ	89.	+0.4 μ	+0.2 μ
57.	+1.1	+0.1	68.	+0.9	+0.0	79.	−0.3	+0.4	90.	−0.1	+0.3
58.	+0.9	+0.1	69.	+0.6	+0.6	80.	−0.2	+0.3	91.	−0.2	+0.3
59.	+0.8	+0.2	70.	+1.1	+0.0	81.	−0.2	+0.4	92.	+0.4	+0.3
60.	+0.2	+0.0	71.	+0.7	+0.1	82.	+0.2	+0.4	93.	+0.0	+0.2
61.	−0.2	+0.2	72.	+0.1	+0.2	83.	+0.4	+0.4	94.	+0.8	+0.2
62.	+0.0	+0.3	73.	+0.1	+0.4	84.	+0.5	+0.5	95.	+0.3	+0.4
63.	−0.2	+0.3	74.	−0.1	+0.4	85.	−0.3	+0.3	96.	+0.2	+0.1
64.	+0.3	+0.3	75.	+0.2	+0.6	86.	+0.2	+0.4	97.	+0.6	+0.1
65.	−0.6	+0.2	76.	−0.4	+0.5	87.	+0.4	+0.4	98.	+0.3	−0.1
66.	+0.4	+0.2	77.	+0.0	+0.4	88.	+0.7	+0.4	99.	−0.8	+0.0

	(*a*)	(*b*)
100.	+0.0	+0.0

It will be seen from this table that slight systematic deviations between the two sets of results exist, but they fall in value within the limits of the accidental errors of observation.

CORRECTIONS TO THE TENTH OF INCH SUBDIVISIONS.

From the data derived from observation, five combinations are made in this discussion. The combination for each series is indicated by the reference numbers given.

The values of $(R - L)$ are given in divisions of the micrometer, one division of which is equal to 0.203 μ = 0.000008 inch.

Combinations.

(1) Subdivisions into 40 equal parts.

(2) Subdivision into 2 equal parts, and the subsequent subdivision of each of the 2 parts into 20 equal parts.

(3) Subdivision into 4 equal parts, and the subsequent subdivision of each of the 4 parts into 10 equal parts.

(4) Subdivision into 5 equal parts, and the subsequent subdivision of each of the 5 parts into 8 equal parts.

(5) Subdivision into 8 equal parts, and the subsequent subdivision of each of the 8 parts into 5 equal parts.

DATA FROM OBSERVATION.

(1) 2-inch spaces.

	$R-L$		Σ
1.	+10.0	+8.0	+8.0
2.	− 6.0	−8.0	+0.0
Mean:	+ 2.0		

(2) Inch spaces.

	$R-L$		Σ
1.	+ 9.0	+0.0	+0.0
2.	+17.0	+8.0	+8.0
3.	+ 1.5	−7.5	+0.5
4.	+ 9.5	+0.5	+0.0
Mean:	+ 9.0		

(3) ½-inch spaces.

	$R-L$		Σ
1.	+7.5	+3.7	+3.7
2.	+1.0	−2.8	+0.9
3.	+7.0	+3.2	+4.1
4.	+8.5	+4.7	+8.8
5.	−5.0	−8.8	+0.0
6.	+4.2	+0.4	+0.4
7.	+0.8	−3.0	−2.6
8.	+6.5	+2.7	+0.0

(4) ¼-inch spaces.

	1 *in.*	2 *in.*	3 *in.*	4 *in.*
1.	+7.0	+13.0	+ 4.0	+8.0
2.	+9.0	+ 3.0	+ 6.0	+6.0
3.	+8.0	+ 8.0	+ 2.0	+8.0
4.	+5.0	+ 3.0	+13.0	+6.0
5.	+6.0	+13.0	+ 1.0	+9.0
Means	+7.0	+ 8.0	+ 5.2	+7.4

(5) $\frac{1}{10}$-inch spaces.

	½ *in.*	1 *in.*	1½ *in.*	2 *in.*	2½ *in.*	3 *in.*	3½ *in.*	4 *in.*
1.	+ 7.0	+9.0	+ 9.0	+10.0	+ 8.0	+ 8.0	+9.0	+10.0
2.	+ 9.0	+9.0	+10.0	+ 9.0	+ 7.0	+12.0	+8.0	+11.0
3.	+10.0	+7.0	+ 9.0	+ 6.0	+10.0	+10.0	+8.0	+ 6.0
4.	+ 9.0	+8.0	+ 6.0	+12.0	+ 8.0	+ 6.0	+7.0	+12.0
5.	+11.0	+7.0	+10.0	+10.0	+ 4.0	+ 6.0	+8.0	+ 9.0
	+ 9.2	+8.0	+ 8.8	+ 9.4	+ 7.4	+ 8.4	+8.0	+ 9.2
		Means: +8.85				Means: +8.25		

I. From (1) and the two halves of (5).

	(*a*)	(*b*)	*Sum.*
1.	+0.4	−1.8	−1.4
2.	+0.8	−1.7	−0.9
3.	+1.2	−0.6	+0.6
4.	+1.6	−0.4	+1.2
5.	+2.0	+1.8	+3.8
6.	+2.4	+1.9	+4.3
7.	+2.8	+2.0	+4.8
8.	+3.2	+0.2	+3.4
9.	+3.6	−0.6	+3.0
10.	+4.0	−2.5	+1.5
11.	+4.4	−2.4	+2.0
12.	+4.8	−1.2	+3.6
13.	+5.2	−1.0	+4.2
14.	+5.6	−3.9	+1.7
15.	+6.0	−2.8	+3.2
16.	+6.4	−1.6	+4.8
17.	+6.8	−1.4	+5.4
18.	+7.2	−4.3	+2.9
19.	+7.6	−1.2	+6.4
20.	[+8.0]	+0.0	+8.0
21.	+7.6	−0.2	+7.4
22.	+7.2	−1.5	+5.7
23.	+6.8	+0.3	+7.1
24.	+6.4	+0.0	+6.4
25.	+6.0	−4.2	+1.8
26.	+5.6	−4.5	+1.1

II. From (2) and (5).

	(*a*)	(*b*)	*Sum.*
1.	+0.0	−1.6	−1.6
2.	+0.0	−1.2	−1.2
3.	+0.0	+0.2	+0.2
4.	+0.0	+0.6	+0.6
5.	+0.0	+3.0	+3.0
6.	+0.0	+3.4	+3.4
7.	+0.0	+3.8	+3.8
8.	+0.0	+2.2	+2.2
9.	+0.0	+1.6	+1.6
10.	[+0.0]	+0.0	+0.0
11.	+0.8	−0.1	+0.7
12.	+1.6	+0.8	+2.4
13.	+2.4	+0.7	+3.1
14.	+3.2	−2.4	+0.8
15.	+4.0	−1.5	+2.5
16.	+4.8	−0.6	+4.2
17.	+5.6	−0.7	+4.9
18.	+6.4	−3.8	+2.6
19.	+7.2	−0.9	+6.3
20.	[+8.0]	+0.0	+8.0
21.	+7.2	+0.1	+7.3
22.	+6.4	−0.8	+5.6
23.	+5.7	+1.3	+7.0
24.	+4.9	+1.4	+6.3
25.	+4.2	−2.5	+1.7
26.	+3.4	−2.4	+1.0

III. From (3) and (4).

	(*a*)	(*b*)	*Sum.*
1.	+0.7	−2.2	−1.5
2.	+1.5	−2.4	−0.9
3.	+2.2	−1.6	+0.6
4.	+3.0	−1.8	+1.7
5.	[+3.7]	+0.0	+3.2
6.	+3.2	+1.0	+4.2
7.	+3.6	+2.0	+5.6
8.	+2.1	+1.0	+3.1
9.	+1.5	+1.0	+2.5
10.	[+0.9]	+0.0	+0.9
11.	+1.5	+0.2	+1.7
12.	+2.1	+1.4	+3.5
13.	+2.7	+1.6	+4.3
14.	+3.4	−1.2	+2.2
15.	[+4.1]	+0.0	+4.1
16.	+5.0	+0.6	+5.7
17.	+6.1	+0.2	+6.3
18.	+7.0	−3.2	+3.8
19.	+7.9	−0.6	+7.6
20.	[+8.8]	+0.0	+8.8
21.	+7.0	+0.6	+7.6
22.	+5.3	+0.2	+5.5
23.	+3.5	+2.8	+6.3
24.	+1.7	+3.4	+5.1
25.	[+0.0]	+0.0	+0.0
26.	+0.1	−0.4	−0.3

	I. From (1) and the two halves of (5).			II. From (2) and (5).			III. From (3) and (4).		
	(a)	(b)	Sum.	(a)	(b)	Sum.	(a)	(b)	Sum.
27.	+5.2	−0.8	+4.4	+2.7	+1.7	+4.4	+0.2	+3.2	+3.4
28.	+4.8	+1.0	+5.8	+1.9	+3.8	+5.7	+0.3	+4.8	+5.1
29.	+4.4	−1.2	+3.2	+1.2	+1.9	+3.1	+0.3	+2.4	+2.7
30.	+4.0	−3.5	+0.5	[+0.5]	+0.0	+0.5	[+0.4]	+0.0	+0.4
31.	+3.6	−2.8	+0.8	+0.5	+0.4	+0.9	−0.2	+1.0	+0.8
32.	+3.2	−3.0	+0.2	+0.4	−0.2	+0.2	−0.8	+1.0	+0.2
33.	+2.8	−3.2	−0.4	+0.4	−0.8	−0.4	−1.4	+0.0	−1.4
34.	+2.4	−4.5	−2.1	+0.3	−2.4	−2.1	−2.0	+0.0	−2.0
35.	+2.0	−4.8	−2.8	+0.3	−3.0	−2.7	[−2.6]	+0.0	−2.6
36.	+1.6	−3.0	−1.4	+0.2	−1.6	−1.4	−2.1	+0.8	−1.3
37.	+1.2	−0.2	+1.0	+0.2	+0.8	+1.0	−1.6	+2.6	+1.0
38.	+0.8	−2.5	−1.7	+0.1	−1.8	−1.7	−1.1	−0.6	−1.7
39.	+0.4	−0.8	−0.4	+0.1	−0.4	−0.3	−0.5	+0.2	−0.3
40.	+0.0	+0.0	+0.0	+0.0	+0.0	+0.0	+0.0	+0.0	+0.0

	IV. From (5).	V. From (2) and (4).			Means I.-V.	Average deviation from mean.	Means I.-V. in microns.
		(a)	(b)	Sum.			
1.	−1.5	.....	.....	.....	−1.5	0.1	−0.30 μ
2.	−1.1	+0.0	+0.0	+0.0	−0.8	0.3	−0.16
3.	+0.3	.....	.....	.....	+0.4	0.2	+0.08
4.	+0.8	+0.0	+2.0	+2.0	+1.2	0.4	+0.24
5.	+3.3	.....	.....	.....	+3.4	0.3	+0.68
6.	+3.7	+0.0	+3.0	+3.0	+3.7	0.4	+0.74
7.	+4.2	.....	.....	.....	+4.6	0.6	+0.92
8.	+2.6	+0.0	+1.0	+1.0	+2.5	0.7	+0.50
9.	+2.0	.....	.....	.....	+2.3	0.5	+0.46
10.	+0.5	[+0.0]	+0.0	+0.0	+0.6	0.5	+0.12
11.	+1.0	.....	.....	.....	+1.3	0.4	+0.26
12.	+2.4	+1.6	+5.0	+6.6	+3.7	1.2	+0.74
13.	+2.9	.....	.....	.....	+3.6	0.6	+0.72
14.	+0.3	+3.2	+0.0	+3.2	+1.6	0.9	+0.32
15.	+1.8	.....	.....	.....	+2.9	0.8	+0.58
16.	+3.2	+4.8	+0.0	+4.8	+4.5	0.7	+0.90
17.	+3.7	.....	.....	.....	+5.1	0.8	+1.02
18.	+1.1	+7.2	−5.0	+2.2	+2.5	0.7	+0.50
19.	+4.6	.....	.....	.....	+6.2	0.8	+1.24
20.	+6.0	[+8.0]	+0.0	+8.0	+7.8	0.7	+1.56
21.	+5.4	.....	.....	.....	+6.9	0.8	+1.38
22.	+3.7	+6.5	−1.6	+4.9	+5.1	0.6	+1.02
23.	+5.4	.....	.....	.....	+6.4	0.6	+1.28
24.	+4.8	+5.0	−0.4	+4.6	+5.4	0.7	+1.08
25.	+0.3	.....	.....	.....	+0.9	0.8	+0.18
26.	−0.3	+3.5	−3.6	−0.1	+0.3	0.6	+0.06
27.	+3.2	.....	.....	.....	+3.8	0.6	+0.76
28.	+4.6	+2.0	+4.2	+6.2	+5.5	0.5	+1.10
29.	+2.7	.....	.....	.....	+2.8	0.3	+0.56
30.	−0.5	[+0.5]	+0.0	+0.5	+0.3	0.3	+0.06
31.	+0.0	.....	.....	.....	+0.6	0.3	+0.12
32.	−0.6	+0.4	+0.6	+1.0	+0.2	0.3	+0.04
33.	−1.8	.....	.....	.....	−1.0	0.6	−0.20
34.	−2.7	+0.3	−0.8	−0.5	−1.9	0.5	−0.38
35.	−3.3	.....	.....	.....	−2.8	0.2	−0.56
36.	−1.8	+0.2	−0.2	+0.0	−1.2	0.5	−0.24
37.	+0.6	.....	.....	.....	+0.9	0.2	+0.18
38.	−1.9	+0.1	−1.6	−1.5	−1.7	0.1	−0.34
39.	−0.5	.....	.....	.....	−0.4	0.1	−0.08
40.	+0.0	+0.0	+0.0	+0.0	+0.0	0.0	+0.00

It appears from this discussion that the average deviation of the corrections for a single space from the mean of the results derived from the different combinations is 0.1 μ, a result nearly identical with that found from the first discussion. It seems safe to conclude, therefore, that in the summation of ten values of $(R - L)$ no error greater than 0.4 μ is likely to occur, when the comparisons are made with care.

It is to be noted that the close agreement of the results in the direct summation of the millimeter corrections with the mean values, must be considered as accidental, since ordinarily it would be impossible to make 40 summations without introducing systematic deviations of considerable magnitude. It is doubtful if anything is gained by making the summations exceed ten in number.

account a bending strain, which he acknowledges does exist. Using my theory as a rough approximation of the truth, it can be shown that even a greater strain is thrown upon the joint than he suggests.

What are the strains in the rim, supposing it to be relieved of one-half of the hoop tension, and its bending due to the inward pull of the arms is one-half of the bending, if the arms do not stretch at all?

Fig. 36 represents the strains in such a case: S and S are the

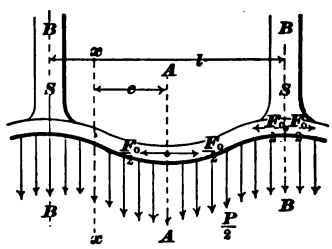

FIG. 36.

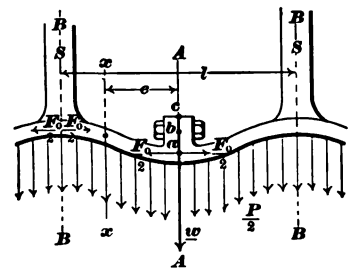

FIG. 37.

arms. The rim is a beam, uniformly loaded and fixed in direction where it joins the arms; it is under tension by an amount $\frac{F_o}{2}$ per square inch.

If $M =$ bending moment at section AA, midway between arms, supposing rim to be uniformly loaded and supported at the ends only; and $M_A =$ bending moment at section AA, supposing rim uniformly loaded and fixed in direction at its ends; and $M_B =$ bending moment at section BB, supposing rim uniformly loaded and fixed in direction at its ends, then $M_B = M - M_A$, and if rim is of uniform section $M_A = \frac{1}{2} M_B$.

The rim should be thick enough to resist both the bending moment M_B, and the hoop tension; it can be determined by the following formula:

$$t = \frac{.475\, d}{N^2 \left(\frac{F}{V^2} - \frac{1}{10}\right)} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

as shown in my former paper; where $t =$ thickness in inches for rectangular rims, $d =$ diameter of wheel in inches, $N =$ number of arms, $V =$ velocity of rim in feet per second, $F =$ maximum working tensile strength of material.

If a joint is used, as shown in Fig. 37, the same rim thickness, t, can be retained, provided the joint is as strong as the solid section of the rim, with a slight additional allowance for the strain caused by the centrifugal force of the lug flanges and their bolts.

What conditions are necessary to make the joint as strong as the solid section of the rim?

First. The bolts must be strong enough and have sufficient leverage, bc, to resist the bending moment (pointed out by Professor Lanza).

$$M_F = \frac{F_o}{2} ac = \frac{V^2 A}{20} ac.$$

Second. The bolts and their leverage must resist the bending moment due centrifugal force developed in rim.

$$M_A = \frac{4}{10} \quad \frac{Ad\,V^2}{N^2}.$$

Third. The bolts and their leverage must also resist the bending action at section AA, Fig. 37, due to centrifugal force developed by weight of lug flanges and their bolts.

$$M_L = \frac{3w\ V^2}{10\ N}; \text{ so that}$$

$$Sbc = M_F + M_A + M_L \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

where $S =$ tensile strain thrown upon all the bolts in one joint.

This reduces to the following practical form

$$Sbc = \frac{V^2}{20}\left(\frac{8\ Ad}{N^2} + \frac{6\ w}{N} + A\ ac\right) \quad . \quad . \quad . \quad . \quad (3)$$

V, d, and N have values already given, $A =$ area of rim cross-section in square inches, and $w =$ weight of one set of lug flanges and their bolts in pounds.

It is worth noting that the joint is made stronger with a given size of bolts when bc is as large and ab as small as possible (Fig. 40).

In the construction of the joint care should be bestowed upon

the flange thicknesses of the lugs, especially upon the section taken on the line of bolt holes XY (Fig. 40), which receives the greatest strain; the thickness db should be such that after deducting the bolt holes this section should have a moment of resistance $= Sbc$. The joint is more apt to fail here than the bolts are to break.

Probably very few joints have been designed to satisfy all of these conditions, and yet few have failed. Indeed, in the case

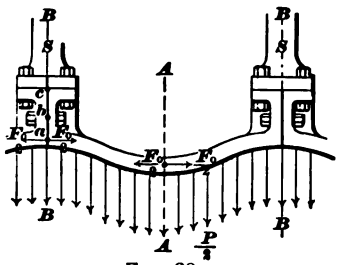

FIG. 38.

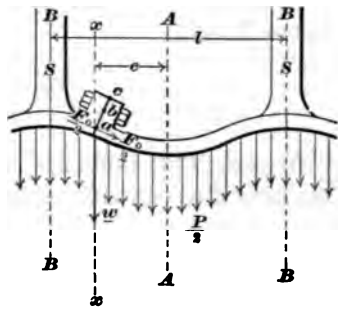

FIG. 39.

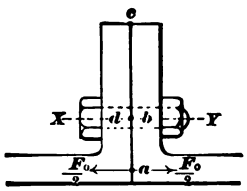

FIG. 40.

of one wheel which burst, the writer is sure the joint did not fail, although not constructed in accordance with these principles. Why? It seems probable that when the bolts are excessively strained they stretch, permitting the joint to gap open slightly at the edge near a, Fig. 37; by this action a greater strain is thrown upon the rim near the arms, so that M_B is made greater. In other words, as M_A is made less M_B becomes larger. If the joint has no strength at all, then Professor Lanza's deduction

$$t = \frac{1.42\, d}{N^2 \left(\frac{F}{V^2} - \frac{1}{10}\right)}$$

is right.

What is the relation existing between strength of joint and thickness of rim?

Let S_1bc be less than Sbc, as determined by equation (3); let

M_X be the available bending moment possessed by the joint, any excess of which will cause it to yield. The bending moment at BB now is $M_B = M - M_X$, M_X being less than M_A.

$M_X = S_1bc - M_F$, and $M = \frac{3}{2}M_A + M_L$, therefore

$$M_B = \frac{3}{2}M_A + M_L - S_1bc + M_F \quad . \quad . \quad . \quad . \quad . \quad (4)$$

In the paper of November last the thickness for rims with joints is taken 50% greater than for solid wheels; this case corresponds to a bending moment $M_B = \frac{3}{2}M_A$ and a hoop tension $\frac{F_o}{2}$; the joint is relied upon to resist the other moments, and it must be strong enough so that $S_1bc = M_L + M_F$, which reduces to $S_1bc = \frac{V^2}{10}\left(\frac{3w}{N} + \frac{A}{2}ac\right)$.

In practice the joint should not be made of less strength than

$$S_1bc = F_oac = \frac{V^2}{10}Aac \quad . \quad . \quad . \quad . \quad . \quad . \quad (5)$$

as pointed out by Professor Lanza, for frequently the edge of a rim may be strained to the full hoop tension.

To calculate a rim thickness for a special joint (whose strength should not be less than that obtained by equation (5)), first determine S_1bc, then by equation (4) find M_B; second, with an assumed cross-section of rim, find the greatest fibre strain F_M induced by bending moment M_B; third, to F_M add $\frac{F_o}{2} = \frac{V^2}{20}$, then $F_M + \frac{F_o}{2}$ must be less than $F = 6{,}000$ lbs. If $F_M + \frac{F_o}{2}$ exceed 6,000 lbs., increase thickness of rim until the desired result is obtained.

Fig. 38 represents the case where the joint is at the arms; the additional strength derived from the bolts connecting with the arms tends to make the joint as strong as the rim itself; the rim also bends in such a manner as to close the joint together near

a, instead of causing it to gap open as in Fig. 37. Therefore the bending moment

$$M_B = 2M_A + M_L - M_F \quad . \quad . \quad . \quad . \quad . \quad . \quad (6)$$

The best position for a joint when it cannot be placed at the arms is that shown by Fig. 39, at a section XX, where there is no bending moment; this section is located a distance $e = 0.289\ l$ from section AA; l = distance between arms.

Under these conditions the joint should be made strong enough to resist the hoop tension $\frac{V^2}{10} Aac$, and the bolts should be proportioned to resist a shearing force equal to one-half the centrifugal force of that portion of the rim included between the joint and the section $AA = \frac{.088 V^2 A}{N}$.

DLXVI.*

EXPANSION BEARINGS FOR BRIDGE SUPERSTRUCTURES.

BY GEORGE S. MORISON, CHICAGO, ILL.
(Member of the Society.)

AMONG the special details of bridge construction, one of the most important is the provision for changes of length due to temperature, and while this is a provision which is nearly always made, it is, unfortunately, one of the details on which least care is bestowed. For many years I have observed the defects of the expansion bearings of bridges as generally constructed, and it has seemed to me that the time has come when the attention of engineers should be called to these defects and to the arrangements which, I believe, will obviate the worst existing difficulties.

When a structure which changes its length rests on supports, the distance between which is practically unchangeable, provision must be made for that change of length. When metallic structures rest on masonry piers some attempt at such provision is universally made. The refinement of modern theory would say that like provision should be made for wooden structures resting on masonry, as timber changes its length under strains more than iron does. In practice, no such provision is made in wooden bridges, and it is best that none should be made. In practice, this provision, though always made in metallic bridges, seldom operates as it should.

The older devices for this purpose were of three classes, sliding plates, rockers, and rollers.

The sliding plate was the simplest of all. It consisted simply of a smooth-planed surface of metal on the masonry, and a like surface at the bottom of the end post, one sliding on the other. So long as the plates could be kept lubricated this worked pretty well; the difficulty in lubricating, however, was often

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fatal, and though there are instances of important bridges, of spans of considerable length (like the deck spans of the Louisville bridge), which have done good service with sliding-plate bearings, the practice has not met with favor. Sliding plates, however, are generally used under riveted structures of moderate length.

Rockers are an attractive design, and have some advantages over other plans. The rocker is, in fact, nothing but a section of a wheel, the weight being carried on the axle and the wheel rolling on its circumference. There are two objections to a rocker bearing; the great weight that must be carried on the axle and the friction on this axle. The friction causes more or less sliding motion; the result of this sliding motion is to work the bearing gradually toward one edge of the section, so that at length the rocker falls over. Rockers were used considerably twenty-five years ago; they are now almost entirely abandoned.

Rollers have always been the favorite device. It is the one arrangement in which friction can virtually be eliminated. There are no sliding surfaces; the top bearing rolls on the rollers, and the rollers roll on the lower bearing. If perfectly constructed the arrangement must always be perfect. Such has not proved to be the fact, and in many instances the motion on the rollers is less than on the sliding plates.

As rollers ought theoretically to give perfect expansion, and fail to do so, it follows that the defects are not in the principle but in the details of design and manufacture. The problem for the engineer is to devise a form of rollers which will always roll.

The function which an expansion joint has to perform is to transfer the weight of a superstructure whose length changes, to a fixed substructure, without exerting on that substructure any other than a purely vertical strain. Whatever the changes in length or the irregularities of workmanship, the weight of the superstructure, and nothing else whatever, should be transferred to the piers below. Horizontal strains should be entirely eliminated, whether those strains be due to changes of dimensions or to irregularities of bearing.

The older styles of rollers failed to meet these conditions and failed to roll; a study of the older rollers will show why they failed.

The old roller bearings differed from the old sliding plates

simply in that a nest of rollers was placed between the two plates. The lower bearing was a flat cast-iron plate on masonry; the upper bearing was the planed surface of an end post, usually of cast iron; the nest of rollers was between. There was no provision except accuracy of workmanship to insure parallelism of the two plates, and even the slight deflection of the truss due to the passage of a train would disturb this parallelism; the necessary result was that some of the rollers were greatly overburdened. This practice, however, was abandoned many years ago, and all the better designed bridges now transfer their weight to the rollers through bolsters, which, supporting the trusses through the end pins, are free to rock; one of the serious defects of the older roller bearings has been thus removed.

In the older bridges built by the Keystone Bridge Company the bottom bearing was a cast-iron wall plate; the rollers were of cast iron; the upper bearing was a cast-iron plate which covered not only the top but the sides and ends of the rollers, shutting them up in a box where they could not be seen. While this arrangement was in many respects the best in use at the time, it had the serious defect of putting the rollers where they could not be kept clean.

A better practice, which was used by Mr. C. Shaler Smith and some other good designers, was to cut a series of parallel grooves in the flat surface of the lower casting; these grooves permitted of cleaning the rollers, and served as dust traps, from which the dust would either blow out or could be removed. It was a step in the right direction, and it is to be regretted that it has not been more generally followed.

The practice which followed this, and which is still in common use among the majority of American bridge builders, places on the masonry a comparatively thin plate of wrought iron or steel, the top surface of which is planed, and on which two angle irons are riveted, which serve as guides for the rollers. The rollers are seldom more than three inches in diameter, and are turned with short journals at the ends, while side plates fitted over the journals keep the rollers parallel, a stiff angle iron frame being sometimes substituted for the plates. Above the rollers is the top bearing, which is the under side of the bolster, and is fitted with guide angles in the same way as the bottom plate. It is impossible to get at the rollers to keep them clean; the guide

angles of the top and bottom plates make a box which becomes filled with dirt, while the rollers are so small that comparatively little obstruction locks them fast. Unless unusual care is taken to secure a proper bearing between the lower plate and the masonry, this thin plate will become indented so that there is a groove under each roller. Bearings of this kind are often found to be completely immovable.

The friction of these small rollers is considerable; if everything were clean it would not be large, but as soon as dirt begins to collect around the rollers friction increases rapidly. The rollers do not always keep in their true position; the side plates of the frames keep them parallel to each other and at equal distances; but the whole nest of rollers will often be found to be badly slued; sometimes the whole nest moves, keeping its proper shape, but more often the guides prevent this, the rollers move individually, and the nest ceases to be rectangular.

The European practice has generally been much better than the American practice; the rollers used have been of larger diameter, and they have been supported on wall plates of greater thickness. The use of large rollers has led to the adoption of segmental rollers, the sides of the rollers being planed off so that they could be put closer together and still as much cylindrical surface as was actually needed for the prescribed motion retained. These segmental rollers are sometimes so designed that they will shut into each other as they roll, but in many cases the sides are simply planed off flat. In Germany they are termed stilts, and their use has become very general.

There seems to be only one valid objection to the use of segmental rollers, and that is that if moved further than is intended they will tip over and it will be very difficult to restore them to position. With very heavy piers this condition is not likely to occur, but in this country, where light piers are sometimes deflected by ice (which can be done without serious injury to the masonry), this accident might not be infrequent. This accident has occurred in Germany, and methods of preventing this overturning have been adopted; they would be doubly necessary here. One method has been to provide the ends of one roller with teeth, which teeth mash into gears in the top and bottom plates; another method is to make the rollers of such shape that they cannot tip over, but will support each other before they fall.

On one important American railroad the engineer has been so dissatisfied with the working of roller bearings that he has given them up entirely, and now secures his expansion by suspending one end of each span by short links from an iron frame that rests on the masonry, the end swinging as it were from a short pendulum. This arrangement has some advantages, but it is compli-

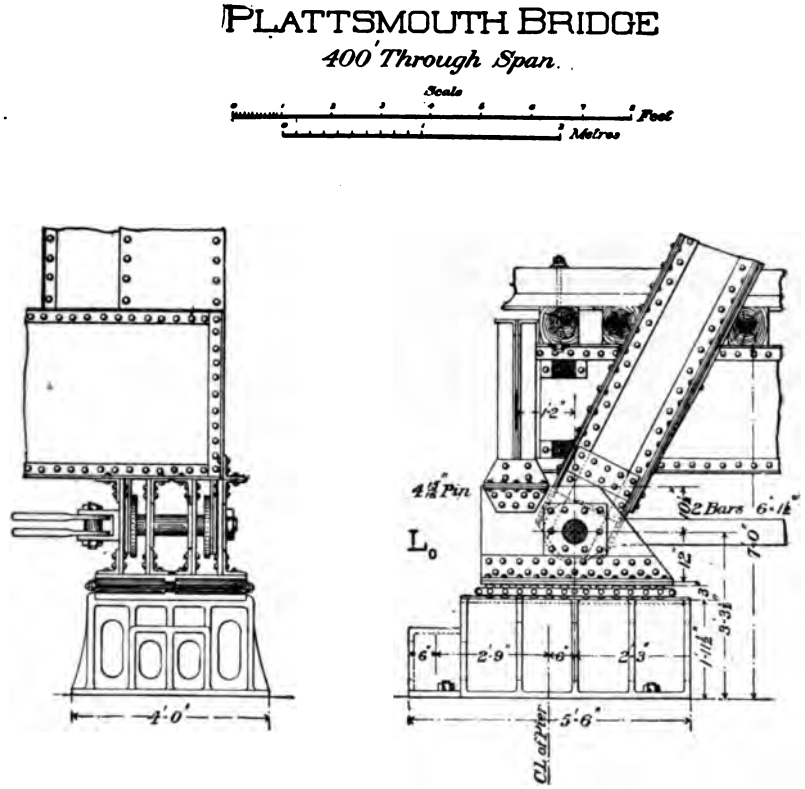


FIG. 41.

cated, and unless the span be anchored extraneously it permits of too much lateral motion.

In 1879, in preparing the plans of the bridge across the Missouri River at Plattsmouth, Nebraska, I used the usual form of rollers running between planed plates, both plates and rollers being of steel, the rollers having grooves cut at the center, and both top and bottom plates having guide ribs on them. The lower plate, instead of resting on the masonry, was supported on

a heavy wall-plate of cast iron. A plan of these rollers is given in Fig. 41. The rollers had the usual journals at the ends, and were kept equally spaced by side plates, which were held together by cross rods with nuts at the end. In order to exclude dirt a piece of rubber belting was placed between the side plates and the rollers, so that when the cross rods were screwed up tight the rubber would form a gasket pressing against both top and bottom plates, the two rubber gaskets, with the two end rollers, forming a box, which was expected to exclude dirt from within. By removing the nuts at the ends of the rods the side plates and gasket could be taken off and everything could be cleaned. The arrangement was not wholly satisfactory; the rubber failed to exclude dust, and after a time was removed entirely; the rollers are now kept clean by occasionally removing the side plates and cleaning them out with a brush. While they work better than rollers generally do, the arrangement is far from perfect.

Two years later, in building the bridge across the Missouri River at Bismarck, N. D., the spans of which are of the same length as those at Plattsmouth, I used the same rollers and the same bearing plate on top, this bearing plate being in both instances the under side of the bolster, but substituted for the flat plate under the rollers a composite plate made of T-rails riveted to a plate below, this plate resting on a large wall plate casting. The arrangement is shown in Fig. 42. One flange of each T-rail was planed to a bevel edge and the other flange planed almost entirely away, the arrangement being such that the flange, when riveted down, held down the side of the adjoining rail; when the whole was riveted up the top was planed off, the center rail being left high, so as to form a guide rib. This gave a very rigid bearing plate, and also one which can easily be cleaned. The dust naturally drops in between the rails, and if it ever becomes necessary to clean the bearing by working a brush parallel to the rollers, the dirt has to be brushed only as far as the opening between two adjoining rails. I have used this form of bearing plate almost invariably since that time, and made no further change in the general arrangement until 1890.

The construction of the bridge across the Mississippi River at Memphis, Tenn., presented a problem which called for a special solution. The central span of this bridge is 621 feet long, and as a cantilever arm projects from each end of this span, the weight carried at each end corresponds to nearly double the weight of

the half span. Making no allowances for changes of length due to strains, the provision for temperature alone required a possible movement of about eight inches, while the actual dead weight carried on each bearing was 2,454,000 lbs., and the total dead and

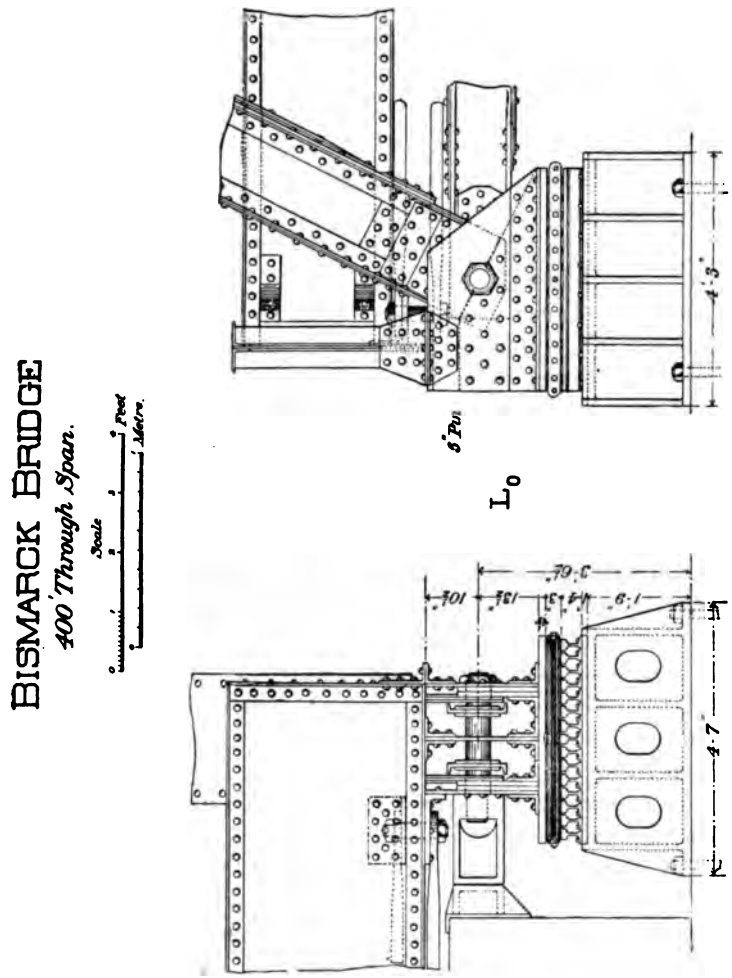


FIG. 42.

live load 4,036,000 lbs. In other words, I had to prepare an expansion bearing with a possible movement of nearly ten inches, which should roll freely under a weight of 2,000 tons. This is probably the largest expansion bearing ever built.

As the piers were tall and slender it seemed necessary to reduce

the friction on this bearing as much as possible, and I decided to use large segmental rollers. The rollers were made of forged steel, fifteen inches in diameter, and placed six inches between centers, the sides being planed out hollow, so that the rollers would shut into each other, after the European practice. This expansion joint is shown in Fig. 43. The under bearing consisted of a wall plate casting made in four sections, the whole being ten feet square. On this wall plate are the rail plates, one on each side of the center line, each rail plate being formed of fifteen rails riveted to a heavy plate below. A solid steel plate four inches thick is bolted between the wall plate castings and extends above the top of the rail plates, thus forming a central guide for the rollers. There are thirty rollers in all, fifteen on each side of the central plate. The sides of the rollers are planed and the cylindrical rolling surfaces are polished. Each roller has two holes bored completely through it, through which pass bolts made of cold rolled shafting, these bolts uniting the rollers on the two sides so that they will roll together. The lower tier of bolts pass through side plates with square ends, and the upper tier of bolts through side plates with hook ends; before the rollers can turn over, the hooks of the upper plates strike the square ends of the lower plates and the motion is stopped. Graduated scales with sliding verniers were fitted on the lower plates, and brass pins were inserted in the hook plates, this being a special refinement put on for the purpose of reading the actual expansion, and from this reading some interesting results have already been obtained.

Above the rollers is the bolster, which it was originally proposed to make of two large steel castings with a guide plate between, like those between the castings of the wall plate: so much difficulty was experienced, however, in getting sound steel castings, that the plan was changed and the bolster was built up of steel plates and I-beams, a central plate being so inserted as to make a guide rib between the rollers. On this was placed a large steel casting, which carries the fourteen-inch pin on which the entire weight of the superstructure is thrown.

As an extra precaution a steel lock plate was placed between the two sets of rollers; before any great strain can be put on the bolts which fasten the side plates to the rollers the bolster will strike one end or the other of this lock plate, and all further movement will cease; it is not likely that this will ever occur, but it was thought best to provide for it, so that in case of some

special strain tending to move the pier, the pier will bend before there is any danger of the expansion bearing being thrown down.

The result of this arrangement has been satisfactory. The

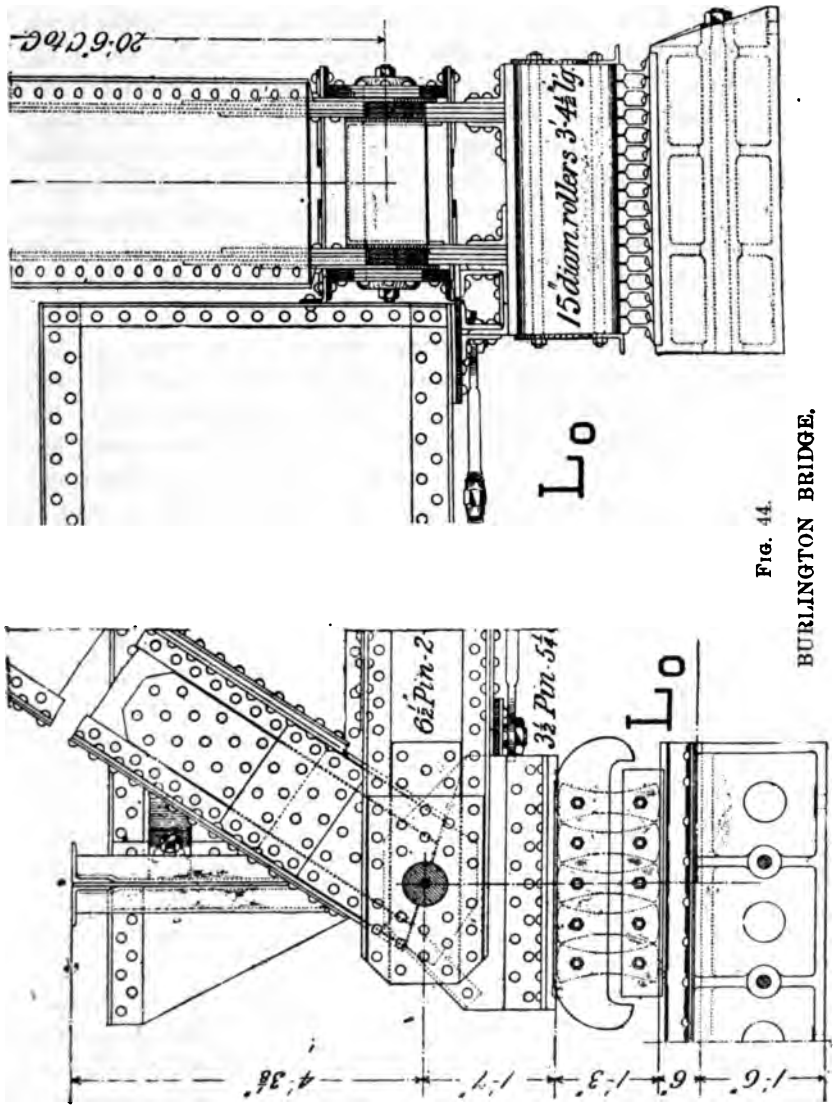


FIG. 44.

BURLINGTON BRIDGE.

bearing is sensitive to all changes of length, it forms an accurate metallic thermometer 620 feet long, and the movements due to the

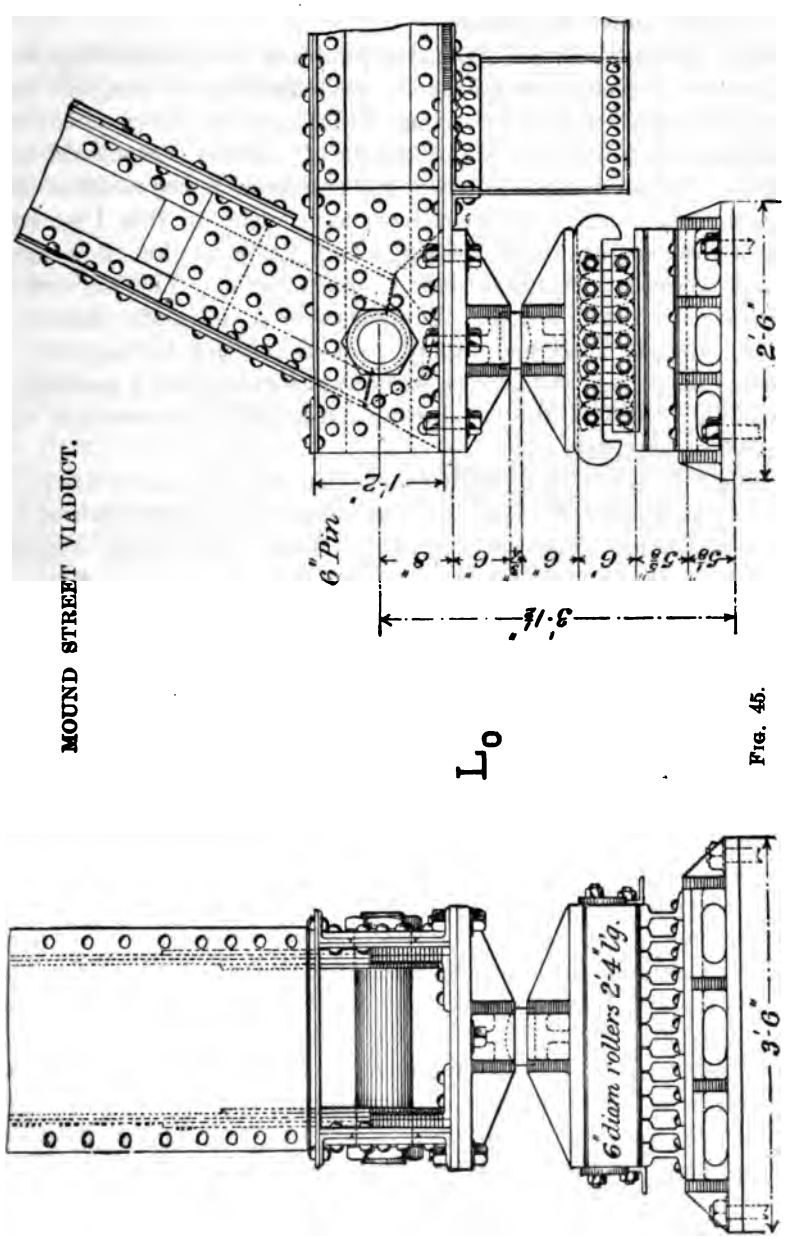
passage of trains and the changes of chord lengths under strains are accurately measured.

The arrangement as worked out proved so satisfactory that I determined to use the same kind of rollers on the bridge over the Mississippi River which I was then building at Burlington, Iowa. This bridge contains six double-track spans, each 246 feet long. The details of the expansion bearings were made precisely like those I had used on other bridges, except that I used large segmental rollers with angle iron side plates with square ends below and flat hook plates above. The side plates projecting above and below the top and bottom bearing plates were made to serve as guides. This bearing is shown in Fig. 44. Each bearing contains five rollers precisely like the thirty rollers at Memphis.

Observations on these bearings showed that some further changes could be adopted to advantage. The cast-iron wall plates and the rail plates were all that could be desired; the large segmental rollers with the hook side plates were good, but could be improved; the pin connection between the bolster and the truss was good as far as it went, but was defective in that it provided for rocking in one direction only, whereas irregularities of workmanship or other disturbances might destroy the parallelism between the pin and the rollers, and thus make very unequal bearings.

To remedy this last defect I decided to make the rocking of the bearing entirely independent of any of the pin connections in the truss, and to put below the chord line of the truss what I have termed a rocker plate, which is a polished steel plate with two cylindrical surfaces on top and bottom at right angles to each other. The plate bearing on the rollers is a steel casting, with a turned socket in the center, in which the rocker plate fits, and above the rocker plate is another steel casting with a turned socket below. The upper casting is, therefore, free to rock in any direction without disturbing the lower casting.

The first bridge in which this device was used was the viaduct which carries Mound Street across the yard of the St. Louis, Keokuk & Northwestern Railroad, in the city of St. Louis. This viaduct consists of a single span of 293 feet. The expansion bearing is shown in Fig. 45. The rollers were 6 inches in diameter and spaced $2\frac{13}{16}$ inches between centres, being much smaller than those used before; in other respects they were similar to the



MOUND STREET VIADUCT.

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FIG. 45.

Memphis rollers. The steel castings with the intervening rocker plate come above the rollers.

The operation of this expansion joint has been satisfactory, but the rollers seemed smaller than it was expedient to use, and further reflection led to the conclusion that there was no good reason why, within reasonable limits, rollers of uniform size should not be used for all lengths of span, the variations being made in the length and number of the rollers; as a result of this, I worked out a standard expansion bearing which I now use for all bridges under which it seems expedient to place rollers. This expansion bearing is given in Fig. 46. It consists of a rail plate similar to those already described, except that the entire top surface is planed off to a uniform level, without any guide strip; under this rail plate would be the usual iron casting, which, however, is not shown on the plan.

The rollers are made twelve inches in diameter and placed six inches between centers, these dimensions being kept constant for all conditions. The section of the rollers has been slightly changed; the sides near the top and bottom being kept parallel, so that instead of shutting into each other, the rollers will strike before turning over, while a hollow is left in each face for working a brush.

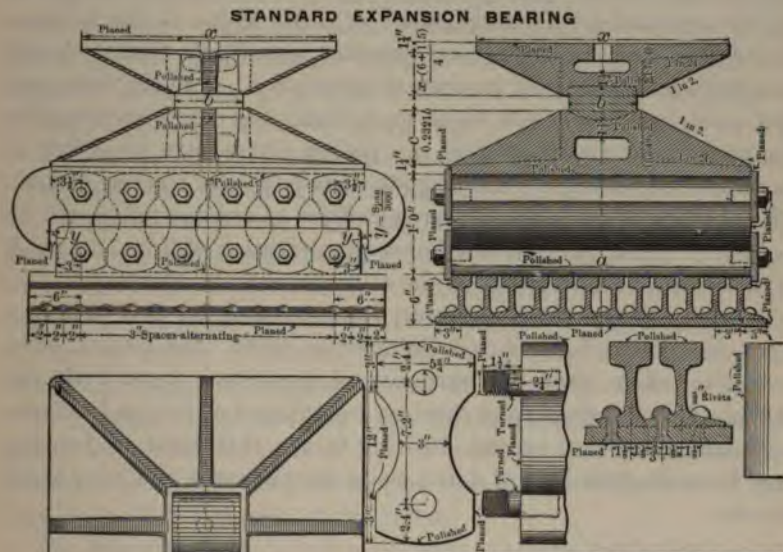
Stud bolts screwed into the ends have been substituted for the long rods passing through holes drilled the whole length of the rollers. The side plates are held by these stud bolts, the lower plate being made square and the upper plate with hook ends. There is thus a double provision against tipping over, the shape of the rollers and the hook plates. Above the rollers comes the top bearing, which is a steel casting carrying the rocker plate which carries the top plate. The lower side plates project below the rollers, and the upper side plates project above the rollers, so that they form guides to keep the rollers in line, taking the place of the guide ribs used in the older bridges.

The bolster or the bottom chord is placed on the top plate, to which it is bolted rigidly. A rocking motion is possible in any direction, and the bearing may be depended on to distribute the weight not only uniformly over the several rollers, but uniformly over the length of each roller.

While not a necessary feature of the design, I have thought it best to make the lengths of the rollers always of multiples of six inches, so that with two rollers we should have practically a bear-

ing twelve inches square, with three rollers a bearing eighteen inches square, etc. I have used this bearing with four rollers under each of the three spans of the new Wabash River bridge on the Ohio & Mississippi Railway. I am using it with ten rollers under the 440 feet double-track spans across the Missouri River at Bellefontaine Bluffs; with six rollers under the 210 feet double-track spans, and with nine rollers under the 360 feet double-track

STANDARD EXPANSION BEARING



No. Rollers	No. Rails	a Length of Rollers Inches	b Side of Rocker Plate Ins.	c $a-(b+1.5)$ / 4 Inches	Total Bearing Inches	Safe load @ 3000lbs.p.in.in Pounds
3	6	17.5	4	3.00	45	135,000
4	8	23.5	5	4.25	80	240,000
5	10	29.5	6	5.50	125	375,000
6	12	35.5	8	6.50	180	540,000
7	14	41.5	9	7.75	245	735,000
8	16	47.5	10	9.00	320	960,000
9	18	53.5	11	10.25	405	1215,000
10	20	59.5	12	11.50	500	1500,000
11	22	65.5	13	12.75	605	1815,000
12	24	71.5	14	14.00	720	2160,000

FIG. 46.

spans of the bridge across the Mississippi River at Alton, Illinois; and with six rollers under the 330 feet single-track spans of the bridge across the Missouri River at Leavenworth, Kansas.

It is possible that, instead of a square bearing plate with two cylindrical surfaces of equal radii (my practice being to make this radius equal to the side of the plate), it would be better to use an oblong plate with different radii at bottom and top, but the only advantage of this would be that it would reduce the bending strain

on the steel castings. Except for this I do not now think of any desirable improvement.

It may be objected that this rocker plate arrangement, which allows a motion in both directions, deprives the end post of a bridge of the transverse rigidity which is due to the usual square bearings. This objection is undoubtedly well taken, and the end posts must either be made stiff enough to resist lateral strains without the aid of square ends, or the two end posts should be rigidly connected by a floor beam or cross frame at the bottom. This latter I regard as the best practice, and think that it should always be followed in all important structures. The rigid connection, however, ought not to be made until the entire structure is completed and swung, so that the rocker plates have been free to move, and so distribute the weights equally on the bearing below; after this has once occurred, irregularities of workmanship will be so largely taken out that any subsequent motion will be too small to be disturbed materially by the stiff connections above.

Contrary to my usual practice, I have patented this arrangement, the patent being dated October 25, 1892. The patent, however, was taken more for purposes of protection than with the expectation of deriving any revenue from it, and all bridge builders, railroads, and other parties desiring to use this form of bearing may have the privilege of doing so on the payment of a very small royalty.

The American practice has generally been to make the permissible weight per lineal inch of the roller proportional to the square root of the diameter of the roller. A common provision in specifications limits the weight in pounds to $500\sqrt{d}$, d being the diameter of the roller. This rule is based on the theory that there is a permissible limit of indentation, which the roller may be allowed to make in the rolling surface, which permissible indentation is supposed not to exceed the elastic limit. If the versed sine of the indentation is constant, the chord of the same indentation will vary as the square root of the radius of the circle; and as the permissible weight is supposed to be a constant of the chord of indentation, which chord varies as the square root of the radius or diameter, the strain is determined accordingly.

While mathematically correct, this rule seems to me based on an incorrect conception of the function of rollers. There is no special reason why the strains either on the surface of the rollers

or on the bearing surface should be kept within the elastic limit; on the other hand, it would be better to have them exceed the elastic limit, if the result of such excess was to reduce the surfaces both of the rollers and the bearing plates to the condition of the surface of cold rolled iron. The only necessary limit is that the elastic limit must not be so greatly exceeded that the metal under the surface will flow, and the roller become permanently deformed, as has occurred in conical roller bearings under turn-table centers.

On the other hand, the same absolute indentation which would lock a small roller so that it would lie stationary in a groove might do little harm to a large roller. If we assume the roller to be set in a groove, we have forces applied on a bent lever tending to lift that roller out of the groove, the vertical arm of the bent lever being the radius of the roller, and the horizontal arm half the width of the groove. So long as the width and depth of the groove are kept proportionate to the radius or diameter of the roller, the force required to roll the roller out of the groove will be constant. In other words, the formula which will maintain for all sizes of rollers a constant freedom of motion, would make the strains on those rollers proportional to their diameters.

In Germany this rule has been followed, and while there has been a considerable variation in the unit strain allowed, the fact has been accepted that the weight placed on a roller should vary directly as the diameter of the roller.

The American rule is not only incorrect in principle but vicious in its results. It encourages small instead of large rollers, a twelve-inch roller, for example, being allowed to carry only twice the weight that would be put on a three-inch roller, so that the smaller the diameter of the rollers the greater the weight which would be permitted on the same area of bearing surface.

In proportioning expansion rollers, I have adopted a somewhat arbitrary basis, accepting 500 $\sqrt{d}$ as correct for a four-inch roller, thus making the weight permissible per lineal inch 1,000 lbs. for a roller four inches in diameter. Instead of varying the weight with the square root, I vary it with the diameter; this makes the permissible weight 250 lbs. per lineal inch on a one-inch roller, and 3,000 lbs. per lineal inch on a twelve-inch roller. The table of weights allowed on bearings of different sizes, which is shown with Fig. 46, has been calculated on this basis.

Herewith is appended an illustration (Fig. 72) showing these bearings.

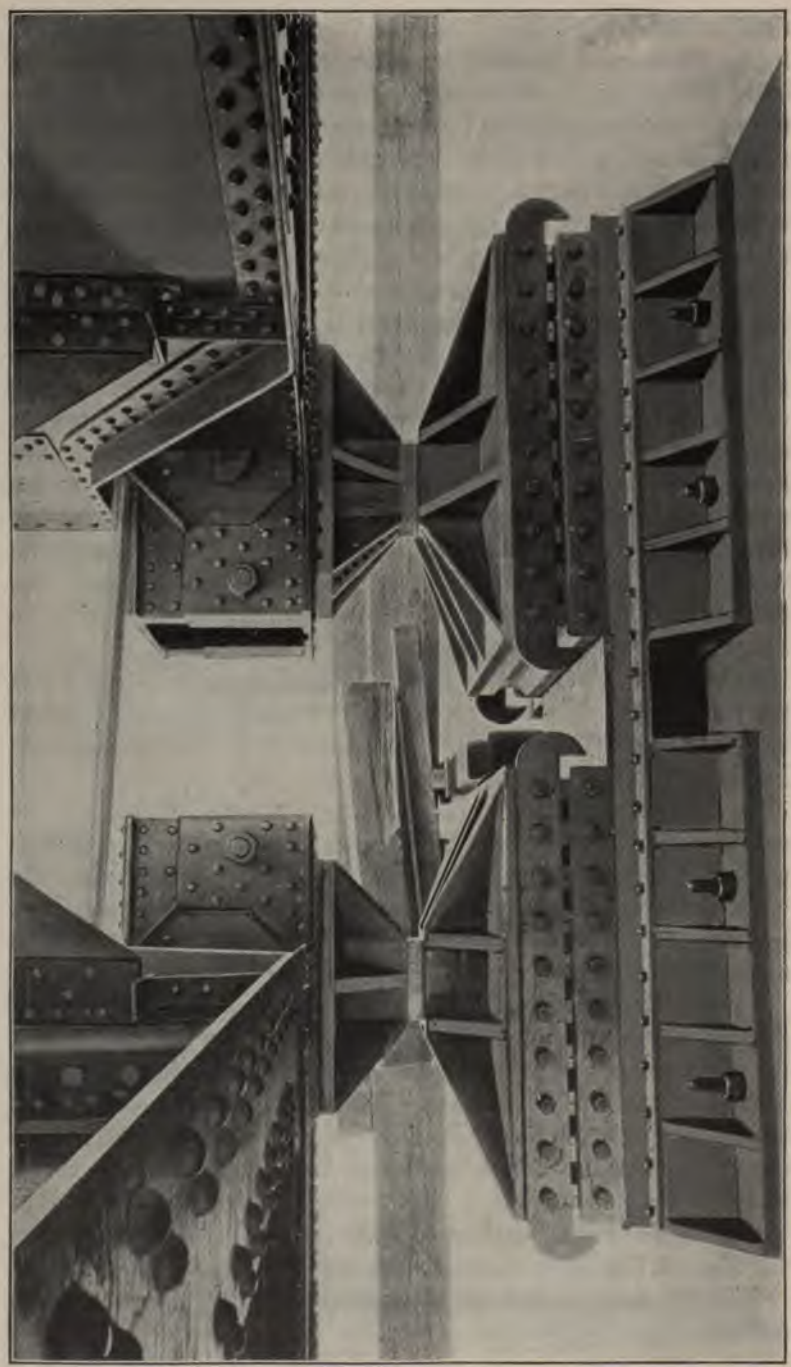


FIG. 72.

DISCUSSION.

Mr. Gus. C. Henning.—As the paper under discussion is one of great importance, I desire to supplement it by a few facts which have come to my notice while observing bridges in operation. It is not generally recognized how much motion there may be at the free end of a bridge span, and how incessant this motion is.

Motion at the roller end of a single-track bridge span, which is the simplest for present consideration, may be caused by the following:

1. Expansion due to temperature;
2. Expansion due to passing load;
3. Lateral flexure due to wind pressure;
4. Lateral flexure due to change of position of sun, or to clouds;
5. Deflection of masonry (or metal) towers due to thrust of an approaching train, ice pressure, or an oblique water current.

Each and all of these produce measurable changes of position of the roller-bearings with reference to the top of piers, and hence cause or tend to cause motion at one or both ends of a span, which may have very serious effects on the stability of the bridge, and permanency of masonry or joints of a metal tower, if proper provision has not been made to avoid its sometimes dangerous effects.

Temperature changes cause regular determinate elongations which are steady.

Passing loads produce not only a determinate elongation, but a strongly marked vibration, which depends largely upon speed of moving load or impact of driving wheels, both vertically and laterally. This is partly due to unbalanced condition of drivers and running gear, but mainly due to irregularity of alignment of track and consequent lateral impact.

Lateral deflection due to wind pressure is of very great importance in long spans, causing a marked difference in length of span, and also a difference in length between the two bottom chords of a long span, hence causing motion at the free end of a span, and skewing the ends. When a bridge is located so that its axis lies in a direction more or less nearly north and south, the passage of the sun from East in morning to West in

afternoon produces a marked difference in length between opposite chords, which causes an alternate skewing of ends of spans, which is relieved by the passage of a heavy cloud, if it takes but five minutes. Such a small interval of time in summer weather allows a sufficient equalization of temperatures in a metal span, on account of rapid radiation, to allow it to return to its normal shape, causing motion at the roller end.

As the greater number of large American bridges have heavy grades on approaches at either end, with the longest spans in the center, the rapid approach of a train, necessary to carry it over the grade, produces a very marked thrust on the towers in the direction of motion of a train. This thrust causes a measurable deflection of masonry piers when they are of the usual American dimensions, and about 100 feet above mean high water and 50 feet below, as is the case in Ohio, Missouri, Mississippi, Hudson River, and other bridges. This deflection is not at all injurious to the masonry when properly built and bonded, but is merely an elastic flexure of the pier, considered as a beam fixed at one end and loaded at the other. This bending is caused even when all the spans rest on roller bearings at one end, and may be transmitted through spans for a long distance, on account of the frictional resistance between rollers and bearings.

It will be self-evident that ice pressure, either when gorged or running, may be a very powerful factor in bending masonry piers, even to a horizontal position; but the mere force of water pressure of a rapidly flowing swollen stream, when cross currents are produced by any reason whatever, causes considerable motion of the piers under a movable end of a bridge span.

From the foregoing it will be seen that ample provision for motion of one or both ends of bridge spans is very necessary, and that altogether too little attention has been paid to this point in bridge construction.

A bridge is always in motion when stresses are normal, but when constrained by rigidity of ends, abnormal strains are produced in the truss or else in the pier supporting it, either of which should be prevented, if possible.

In the case of the Henderson Bridge over the Ohio, at Henderson, Ky., I have noticed that changes due to all of these causes occurred during several months of observations, and I give their magnitudes below. In this bridge, located nearly

north and south, there were two 250-foot spans at the south end, then a 525-foot main channel span; then six 250-foot spans at the north end, all on a 75-feet-per-mile grade except the 525-foot span, which was horizontal.

All of the spans were provided with expansion bearings at the up-hill end, and the 525-foot span at its north end, and these consisted of bolsters, with wrought-iron or steel base-plates resting on steel rollers, running on cast-iron pedestals. The nests of rollers were all guided, so that they could not change their positions relatively, and as a matter of fact the expansion bearings all moved under the action of disturbing influences.

An extended series of observations was made during July, August, and September, 1885, and measurements made of all motions occurring at the roller end of the 525-foot span, and the adjoining 250-foot span having its expansion bearing also adjoining.

1. It was found that the 525-foot span, as well as the others, were true metallic thermometers, in that they expanded and contracted uniformly with changes of temperature, provided no external influences coexisted to counteract the effect of temperature or vitiate observations. Of course, all observations were made on top of one pier, 100 feet (about) above mean high water, considering the same as a fixed and immovable platform.

2. A passing load invariably produced marked vibrations and gradual uniform extension of both expansion bearings, and amounted to $\frac{5}{16}$ inch when a loaded freight train covered the entire span; the vibrations caused rapid and constant changes of length of about $\frac{1}{16}$ inch as a maximum, when the engine had passed about 35 feet beyond the center of span; but the vibrations of that span continued until the train had entirely left the span.

3. Lateral flexure, due to wind pressure, often caused the windward truss to be "bowed" several inches, while the leeward truss went with it; the whole effect was to advance one roller-bearing beyond the other, sometimes as much as $\frac{1}{8}$ inch, while the entire motion was about $\frac{3}{16}$ inch in the case of a moderate cyclonic storm. Vibration was, however, almost absent in such case.

4. Change of position of sun caused a marked skewing of the expansion-end of the 525-foot span of more than $\frac{1}{8}$ inch. In the

morning the sun rising in the East would expand the north truss considerably more than the south truss, which would advance the north roller-end more than $\frac{1}{8}$ inch beyond the position of the south roller-end, both advancing due to the general rise of temperature of the surrounding air as the sun rose higher. When, at noon, the sun stood in the zenith, the ends of the trusses would again stand even; but as the sun traveled toward the West, the reverse motion would take place, and the south truss would elongate more than the other, and the reverse skewing would occur, but to a lesser extent. Both ends would be moving all the time. The passage of a cumulus cloud, however, would have a marked effect, causing almost immediate contraction of the entire span, and would sometimes return both the roller-ends to their normal condition and position. The radiation from a metal structure is so rapid under the influence of a breeze, that a very short time suffices to equalize temperatures. It must be borne in mind that changes of length of bottom chord produce motion of expansion bearings, while those of top chord do not, but merely cause minute changes in angular position of end-posts.

5. The most surprising and sudden cause of motion at the expansion bearings was, however, the impact of a rapidly approaching train on the heavy grade on the north end of the structure. As this grade was 1,800 feet long and over, the engine-drivers were in the habit of "running for it," *i. e.*, giving the engine a full head of steam and acquiring considerable momentum before striking the grade; as the engines then in use were too light to handle the heavy loads easily which they were called upon to draw. When the engine struck the fourth span from the point of observation at the roller-end of the 525-foot span, or 1,000 feet distant, the four spans, piers and all, would be pushed more than $\frac{3}{16}$ inch in the direction of the motion of the train, and this in spite of the fact that all of these spans were resting on rollers. As the train approached the horizontal span at the point of observation, the pier on which the observations were made would gradually settle back into its vertical position, so that as the engine passed the point of observation, the position of the expansion-end on its pier showed that normal conditions again existed. It will not be necessary to explain the action of ice pressure or effect of a current of water, as these are self-evident, and I had no opportunity of measuring their effect.

The expansion bearing described in the paper, and patented by Mr. Morison, is no doubt the best one that has ever been designed for the purpose of relieving bridge trusses and piers from abnormal strains, and it seems reasonable to go to any expense in order to arrive at so desirable a result. If other bridge details were designed with equal care, much better structures would, as a rule, result, and at least some accidents be avoided.

As an evidence of the failure to do this, I will select a remark on page 166 of the paper, where the author prefers to use a sometimes objectionable feature instead of using the correct method, which he indicates at the same time. He calls attention to the looseness of the lower end or foot of the end-post, which deprives the end-post of transverse rigidity, which is due to the usual square bearings, and points out that the proper remedy is increased stiffness in the end-post itself; but immediately recommends his preference "of riveting a floor-beam or cross-frame between end-posts," which may produce flexure in the side webs of the end-posts, and hence are objectionable. The end-posts should, of course, be amply stiffened by portal-struts or bracing, and should also be strongly connected with each other, or anchored at their bases; but undoubtedly the proper way of designing the lower end of an end-post is to make it sufficiently stiff laterally to obviate any possibility of lateral flexure due to any strains that may occur.

The device shown will, no doubt, give all the necessary initial adjustment, and also allow for all possible kind and amount of motion due to the causes mentioned, without producing undesirable strains, or wear and tear. With this construction there never will be any sticking of ends. Rollers and surfaces can be well oiled, cleaned, if necessary, kept free from flying sand—which is such a serious trouble along the Missouri—and the nest is open to inspection at all times. Moreover, the materials selected are such as will make the coefficient of friction the smallest possible. Ice, and snow as well, will have no effect on the action of this bearing, as they will be forced laterally into the spaces. In the use of the maximum lineal pressure allowable on these rollers, Mr. Morison has also advanced a long step in the right direction to assure free motion of the expansion-end; the pressures shown in the table in Fig. 46 give very low pressures, and make sure that the rollers will indent neither upper bearing plates nor lower rail plate, and makes ample pro-

vision against flattening of rollers, which has occurred in many well-known bridges.

In the case of the Henderson Bridge, the materials and proportions were such that the expansion bearings were very mobile, but it is a question whether corrosion in course of time will not interfere with it materially.

There is, however, one slight defect, it appears to me, in the design of this expansion bearing; it is the small dimensions and consequent high bearing pressures on the rocker plate. Examination of the dimensions and loads given shows that a bearing pressure of 8,500 lbs. per square inch of surface has been allowed in the design of the supposed rocking surfaces. Such very high pressure will force all lubricants from between these surfaces, and will make the surfaces stick together without any possibility of motion between them, as the rolling disks would be compressed 0.00112 inches, while they would be forced into the upper and lower rocker plates, as cast is more compressible than wrought steel. Moreover, this calculation is based on the assumption that the cast-steel rocker plates are solid, while the drawings show large core-holes (page 165) directly under the center of rocking disks, which make that central part of the bearing surfaces yielding, and thus increase the pressure per square inch on the remainder of the bearings.

If these rocker bearings came into play only so long as a partial load is on them, this might not be an important matter; but as these bearings find their principal use or *raison d'être* during adjustment of portal braces and top lateral system—which is properly done only after completion of span as far as regards distribution of all dead loads which will come on the structure—the bearings must be easily movable under the total load of span, and I doubt very much that they will, when designed with dimensions as given, be in that condition. This criticism does not in any manner affect the advisability of using such a design, as such a small matter can easily be changed.

Some notable examples of lack of provision for motion at one end of a span are in existence at the present day, and as such I wish to enumerate the old Louisville, called the Fink, Bridge, in which the main-channel spans gradually crept up-hill about seven inches, until it became necessary to draw them back into a reasonable position. There then was a possibility of the struct-

ure drawing itself off its bearings and being precipitated into the river below.

The old Cincinnati Bridge, built by the Keystone Bridge Company, is another case in which the rollers and their bearing plates are commonly reported to be corroded fast, preventing all possibility of motion. There is an endless number of smaller bridges which I have seen in which there never was any motion, and in which the rollers are firmly bedded in a mass of dirt and rust, so that the spans are rigidly anchored, causing cracking of the pointed masonry joints and expense and attention, which would all be avoided if one end were free to move.

I should like to speak about one point which Mr. Morison brought up in his presentation of his paper, and that is, the unnecessary precaution of limiting the intensity of the load upon the rollers within the elastic limit. If the rollers were originally of cold-rolled steel, then I think his remarks are perfectly correct, that it would be less likely to be compressed or injured by rolling; but, inasmuch as these were rolling under the working of the bridge, which would cause a flattening of the roller just where that cold rolling occurs, and only there, I think that would be just the effect we want to avoid. If the material is not cold rolled, and then the excess load that comes on it were beyond the elastic limit, then it would be cold rolled to that point, and we would have the same trouble which has arisen in so many bridges.

*Mr. George S. Morison.**—There are two points in Mr. Henning's discussion about which something may be said. The first of these relates to the method of stiffening the end-posts to overcome the loss of stiffness due to the round bearing at the bottom, and the second relates to the pressure per square inch on the rocker plate.

As regards the first point, the reason for making the stiff connection between the end-posts and the floor-beam is not only to make up for the loss of stiffness due to the round bearing at the bottom, but also—what I consider an important thing in all bridge designs—to make the superstructure as nearly as possible a single beam. I consider it very important not only to provide against all strains due to weight and to wind pressure, but to overcome the various vibrations, which refined mathe-

* Author's Closure, under the Rules.

matical investigation may ultimately explain, but which are now very imperfectly understood. In other words, stiffness as well as strength must be provided for. The practice of riveting floor-beams to vertical posts was at first objected to on account of the strains produced in the posts by the deflection of the floor-beam; these strains undoubtedly exist, but the good effect of the stiff connection is greater than the harm done by these strains, and this detail is now very generally used. On precisely the same principle, I prefer to connect the inclined end-posts with the floor-beams and give the end panel of the bridge all possible transverse stiffness, and for this reason I prefer to use a detail which may be theoretically somewhat incorrect.

The other point relates to the pressure on the rocker plate, which is admitted to be too great for the preservation of lubricants. In bridges which are under constant care it might be well to increase the size of this bearing and provide for lubrication. In bridges which receive only average care, the lubricating would certainly be omitted, and the cylindrical surfaces would be so large as to diminish the leverage and thereby increase the friction. I have, therefore, thought it best to adopt a comparatively small size and admit a comparatively large pressure, this pressure being about one-half that permitted on the pins in a steel bridge. When the bridge is erected, the rocker plate should be carefully packed in oil, and this oil will permit of the necessary movement required until all parts are brought to a bearing; this is as much as will ordinarily be found necessary.

In bridges of exceptional character special provision must always be made which this detail does not provide for. It might be expedient to substitute spherical surfaces for the cylindrical surfaces of the rocker plate—this would permit the rotary motion, which must exist when one chord expands faster than the other; spherical surfaces, however, are very much more difficult to make than cylindrical surfaces, and this rotary motion is generally so slight that it is safe to omit any special provision for it. There are possible conditions in which a provision for expansion in the direction of the length of the bridge is not enough, and the theoretically perfect arrangement would make one only of the four bearings stationary, and place the rollers of the other three on radial lines centering at this single fixed bearing; fortunately, ordinary bridges are not of such dimensions as to make this provision necessary.

DLXVII.*

THE "BUCKEYE" VALVE-GEAR.

BY ALBERT K. MANSFIELD, SALEM, OHIO.
(Member of the Society.)

THE so-called Buckeye engine first came prominently before the steam-using public at the World's Exposition at Philadelphia, and it was just previous to that time that the design of the engine and its valve-gear had been introduced by Mr. J. W. Thompson. Perhaps the most comprehensive explanation of its valve-gear is that contained in *Modern Steam-Engines*, by Joshua Rose, member of this society; and, although not much may be added to what is therein contained, yet the subject is so susceptible to analysis, that it may not be out of place to bring the matter before this society in a more extended form than may be done in a popular writing like that referred to.

Fig. 47 is a diagram representing the method of operating the valve. The compound rocker arm *A*, is the feature of this combination deserving special notice. Its exceeding ingenuity places it very high among recent improvements in steam-engines.

The main rocker *be* carries, pivoted at or near its midway point *d*, the cut-off rocker *cdf*. It follows that the eccentric *h* of constant eccentricity (that is, which has an adjustment about the shaft, not across it) produces a constant travel of the cut-off valve on the main valve, no matter what the point of cut-off may be. This is not difficult to understand from the figure, and is probably very generally understood.

Fig. 48 is a section through main and cut-off valves and cylinder, and Figs. 49 and 50 are a pair of Zeuner diagrams of the main valve movement, differing from each other, in that the first refers to an engine of short stroke (high speed), having a relatively large per cent. of clearance space, while the second is of a long stroke (slow speed) engine having small per cent. of clearance.

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

FIG. 47.

The accompanying sectional view corresponds to the first of the two diagrams.

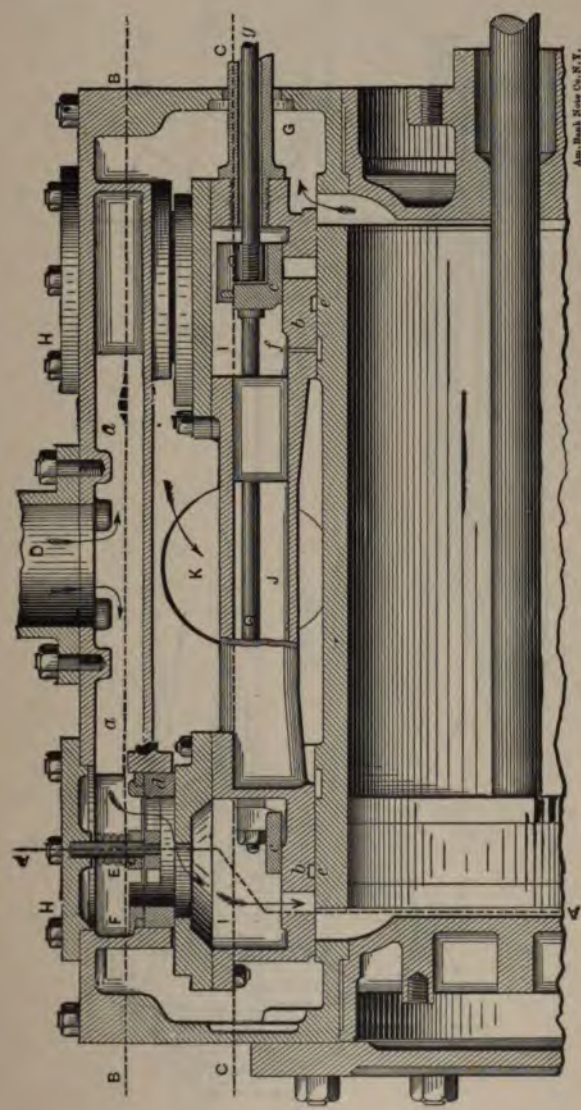


FIG. 48.

These diagrams do not differ from those of any other simple valve movement produced by fixed eccentric, except in the matter of the eccentric following the crank, instead of leading

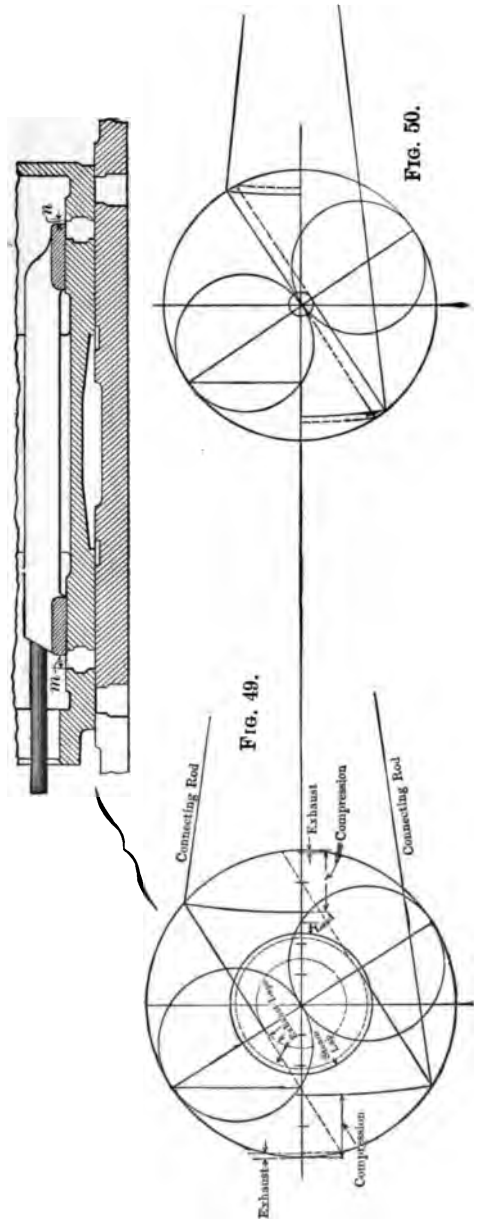


FIG. 49.

FIG. 50.

it. This is due to the hollow shape of the valve, which permits the steam port to be opened to the cylinder by an opposite movement to that which is usual.

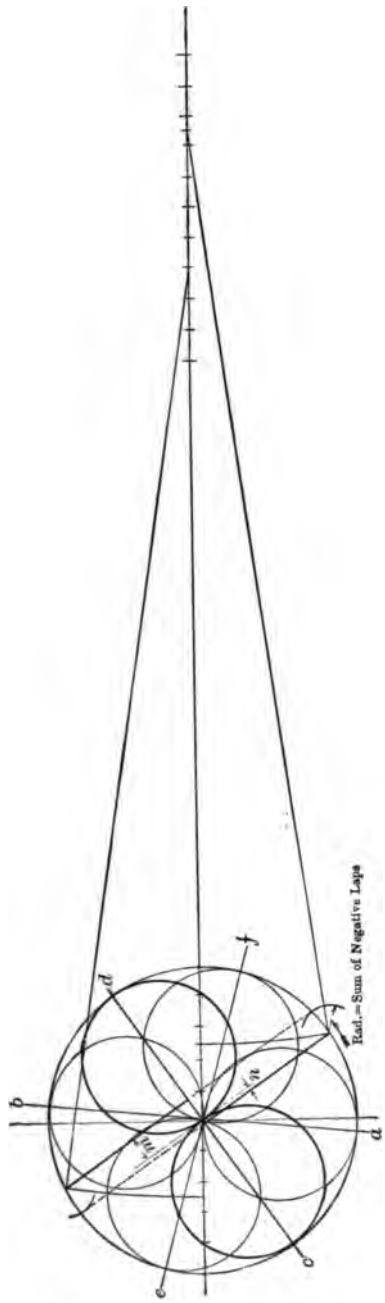


FIG. 51.

It will be observed that the point of release, or exhaust opening, is rather late in the short stroke diagram. This could be improved by increasing the valve travel and the angular advance, but in so doing the main valve does not remain open as long, and therefore the maximum possible power of the engine is decreased. It is found best to compromise these matters in engines having large per cent. of clearance. No such compromise is necessary in the longer stroke engines.

Fig. 51 is a Zeuner diagram of the cut-off valve movement. The heavy circles correspond to one-fourth cut-off, and the four lighter circles to earliest and latest points of cut-off. The range of the governor is usually such that the line *AB* of earliest cut-off is 50° from line *CD* on one side, and line *EF* of latest cut-off is 50° on the other side; or, in other words, the range of automatic action of the cut-off eccentric is usually 100°. The cut-off rocker arm, being a lever of the first order, reverses the valve movement from that of the eccentric. The eccentric is therefore 180° away from the position which the diagram would appear to require.

It will be observed that the position of cut-off eccentric corresponding to one-fourth cut-off is about at right angles to that of the main eccentric. It follows that when cutting off at about one-fourth, at the moment of cut-off the main valve is standing still and the cut-off valve is moving at its maximum speed.

The amount of opening *MM* (Fig. 48) of the cut-off valve, when in its mid position relative to the main valve, may be called the negative lap of cut-off valve. The sum of these negative laps is usually made about the same as the width of the main valve port, for since at its middle position the cut-off valve moves at its maximum velocity; therefore it would be clear that the act of sweeping across the main valve port is done in the least possible time by this adjustment.

Owing to the influence of angularity of the main connecting rod, the negative laps are unequal when cut-offs are adjusted to be alike at both ends of cylinder. It is usual in simple engines to adjust cut-offs at one quarter stroke which corresponds to the normal load.

The method of determining the actual negative laps for each end of the cut-off valve, under the condition named, is indicated in the diagram, and will be clear enough to those who are famil-

iar with the Zeuner diagram. A discussion of the ordinary features of that method of studying the valve movement will be omitted here, it is so completely available elsewhere.

By deriving from the Zeuner diagrams rectangular coördinates to represent the amount of opening of the valve port for different positions of the piston, we form the well-known ellipses of motion as shown in Fig. 52. The shaded portions of this figure give a graphical representation of the variation of this opening

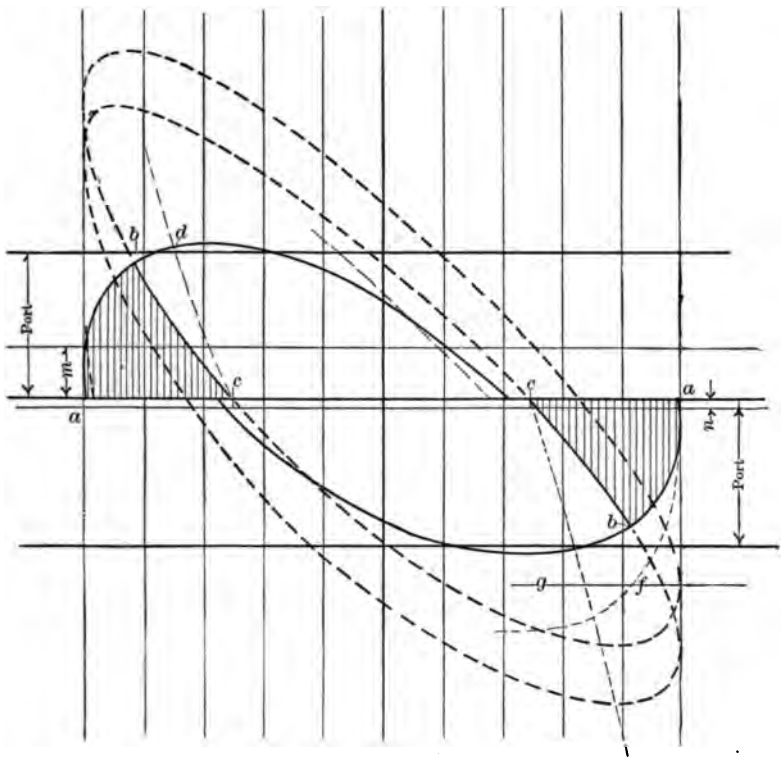


FIG. 52.

as derived from the first part of Fig. 49 and from Fig. 51 when cutting off at one-fourth. These shaded portions ABC will be seen to be bounded by short portions of the main valve ellipses and of the cut-off valve ellipses, intersecting each other. The area of the shaded figures is a measure of the efficiency of the valve opening to deliver steam to the cylinder when cutting off at one-fourth. It is clear that by increasing the valve travel, and

thereby the diagram of the valve movement, this area may be increased. Other methods of increasing the area are, however, available.

Fig. 53 illustrates in the shaded portion the opening and

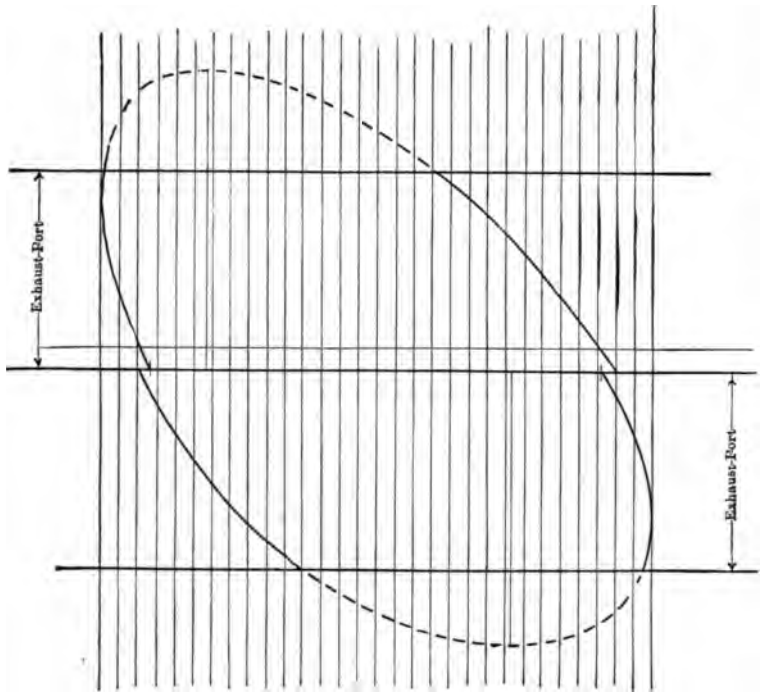


FIG. 53.

closing of the exhaust port, as derived from Fig. *AA*. It will not need further explanation to those familiar with the ellipses of valve motion.

Fig. 54 illustrates a method of applying a double-port action to the cut-off valve, devised by Mr. C. W. Barnaby, and which has been extensively used on Buckeye engines. If the travel of this double-port valve remains the same as that of the simple cut-off valve shown in Fig. 48, then the vertical distances from base line to curve *BC* of Fig. 52 become twice as great as before, which changes this curve to the curve *CD*, thus increasing the area of the figure by about forty per cent. The efficiency of admission is probably not improved by quite this percentage, owing to the somewhat tortuous passage for a part of the steam.

Fig. 55 illustrates a method of double-porting the main valve and quadruple-porting the cut-off. By this method if the main valve travel is three-quarters that of a single ported main valve,

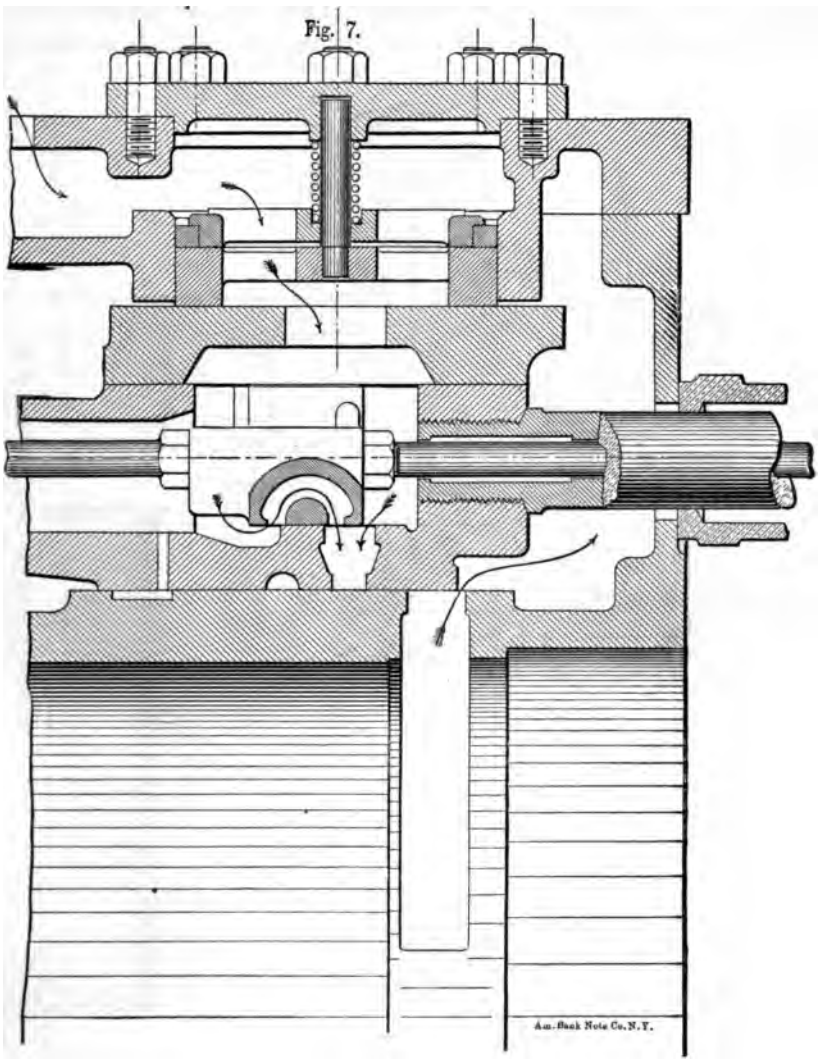
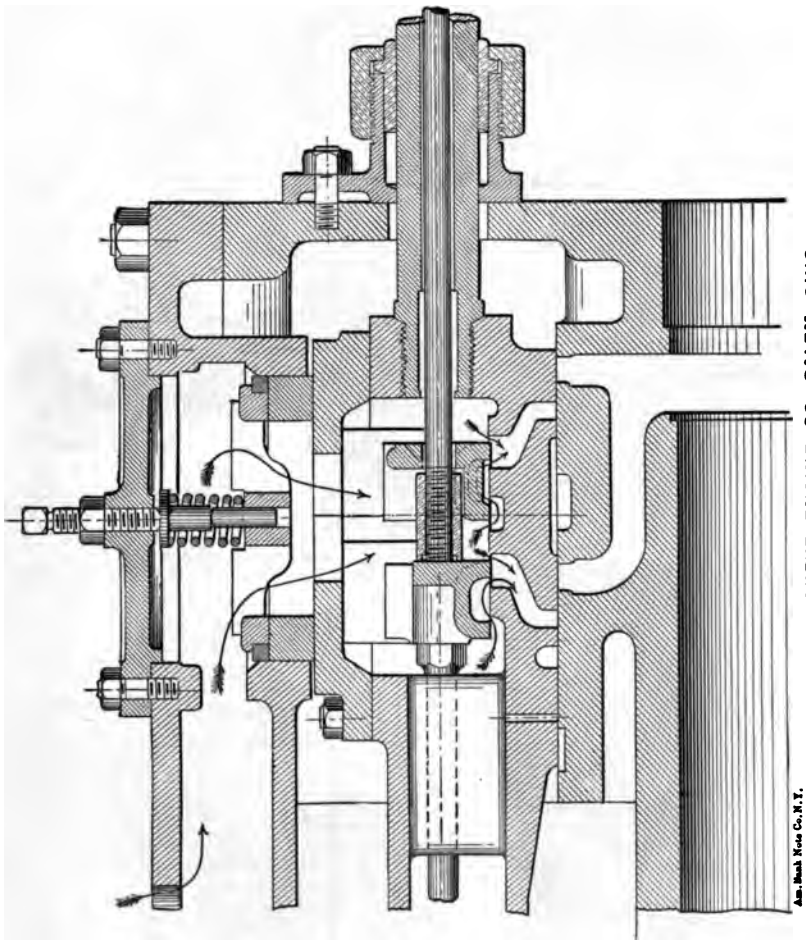


FIG. 54.

then the vertical distances to main valve curve in Fig. 52 become 50% greater than before; and if the cut-off valve travel is three-fourths as much as in the case of a simple cut-off valve, then the vertical distances to cut-off curve become three

times as great as before. Curves *AFGC*, to fulfil these conditions, are drawn in the lower right-hand part of Fig. 52, and the figure therein contained is found to have eighty-six per cent. more area than the shaded figure. This area is taken to the line *FG* only, which is drawn at a distance from the base line of



BUCKEYE ENGINE CO., SALEM, OHIO.

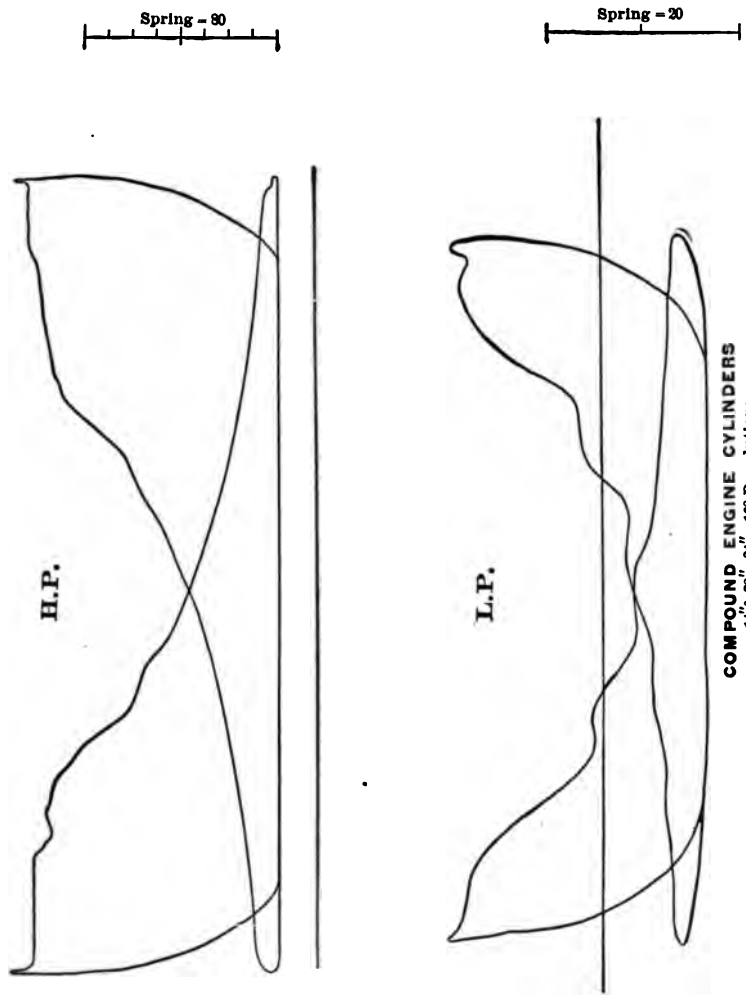
FIG. 55.

the figure equal to the assumed sum of the two main valve ports.

By this device the efficiency of steam admission is increased by eighty-six per cent. less whatever is lost by wire drawing through the multi-ports.

There may be some question as to whether the desired object

is best gained by multi-porting in this way or by increasing the travel of the simple valves to such an extent as to gain the required area of the figure ABC (Fig. 52). On the one hand an increased travel of valve causes an increase of inertia to be over-



COMPOUND ENGINE CYLINDERS
14" & 28" x 24" — 160 Revolutions.
Fig. 56.

come in starting the valve at each end of its stroke, also to an increase in the size of eccentrics. On the other hand, multi-porting the main valve, while decreasing the inertia owing to the decreased travel, increases the cylinder clearance, and makes the parts of the engines somewhat more difficult to produce.

The answer to the question which experience points out is, that for smaller engines the simple construction with liberal travel is best, while for large engines the multi-ported construction may be preferred; especially is this the case with the large cylinders of large multiple cylinder engines. An example of this application was shown in the low-pressure side of the cross-compound condensing Buckeye Engine exhibited at the Columbian Exposition. Fig. 56 shows a pair of cards from this engine.

Fig. 57 shows an adaptation of piston valves to a Buckeye engine. In this arrangement the valve movements are the same relative to each other, and made by the same mechanism as in the case of ordinary Buckeye valves.

Most of the objections which have been urged against piston valves are overcome in this application. The travel of both main and cut-off valves being constant in extent on their seats, the wear is uniform; therefore the valves remain steam tight, where in the case of variable travel this is impossible. Packing rings are easily applied, as shown, at all points where leakage could occur. Only the end surfaces of the main valve are exposed to the current of exhaust steam. The valves are light in weight, and, being relieved almost entirely from steam pressure, the extent of valve travel may be increased without detriment, making it unnecessary to multi-port.

This construction seems from experience to be especially desirable in high-pressure cylinders of large multiple-cylinder engines, where high steam pressures are used. Such an application was made with success in the large triple expansion Buckeye engine at the Columbian Exposition. A set of cards from this engine is shown by Fig. 58, which, while they do not correspond to the best economy of the engine, owing to lack of sufficient load, which had not been available up to this writing, yet they serve very well to illustrate the distribution of steam.

A few indicator diagrams from examples of Buckeye engines are given on pages 196 and 197 to illustrate the capacity of this valve-gear to meet the requirements of various kinds of service. These diagrams are exact reproductions of actual indicator diagrams taken from engines in service. Data relating to the engines are marked on the diagrams, so that further comment will be unnecessary.

The fact need not be disguised that all indicator cards from

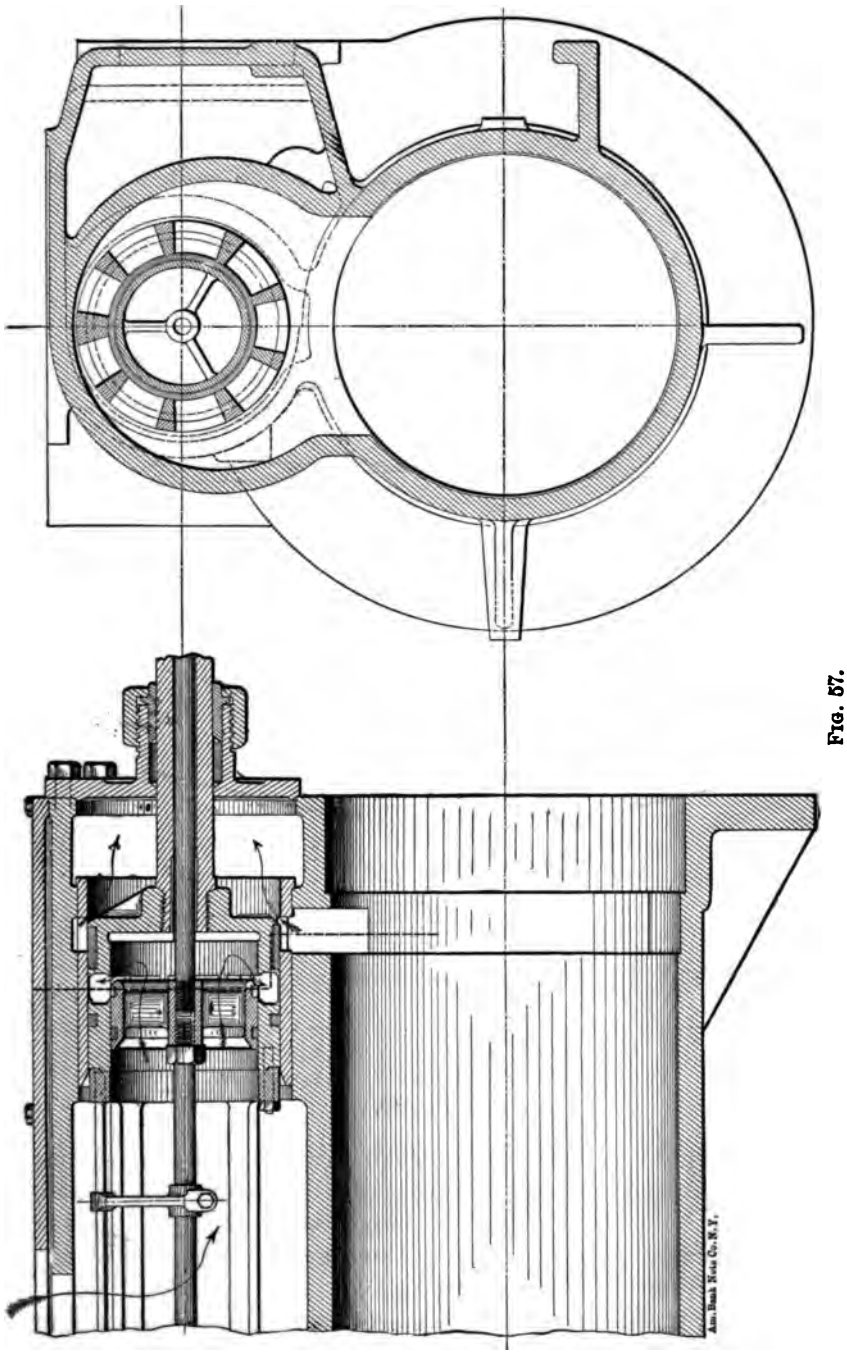
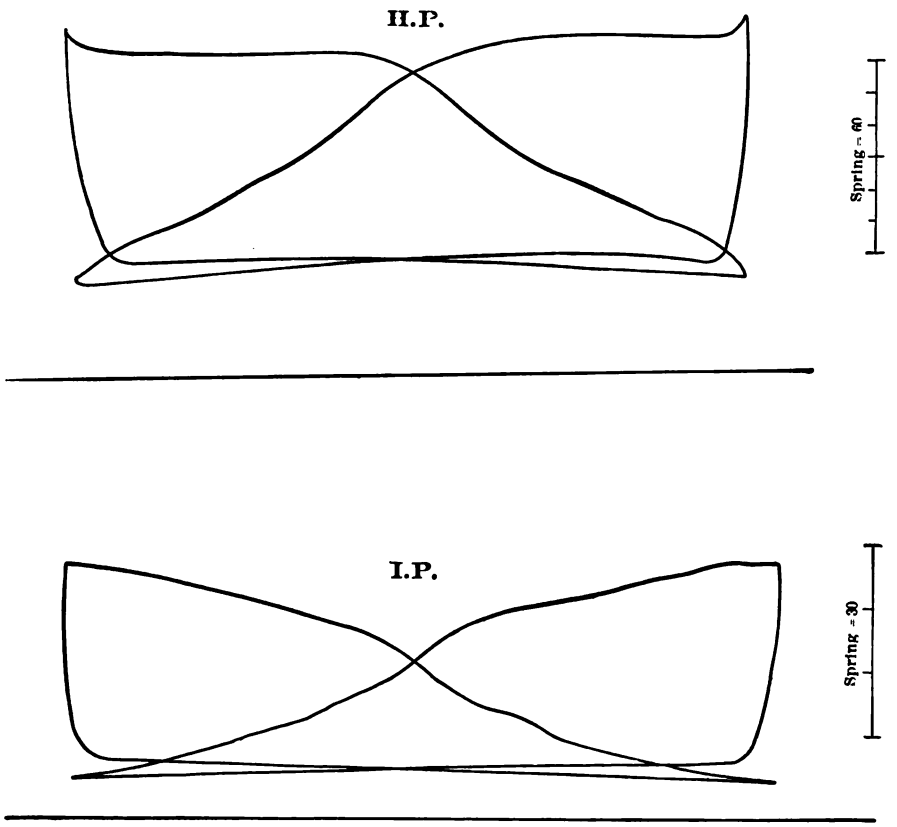


FIG. 57.

this or any other type of engines are not equal to the best. This may be often accounted for by the fact that steam is not properly delivered to the engine from the boiler.

Long, tortuous, and contracted steam-pipes not only injure the shape of the diagram, but even reduce, sometimes to a great extent, the maximum power of the engine as well as its economy.



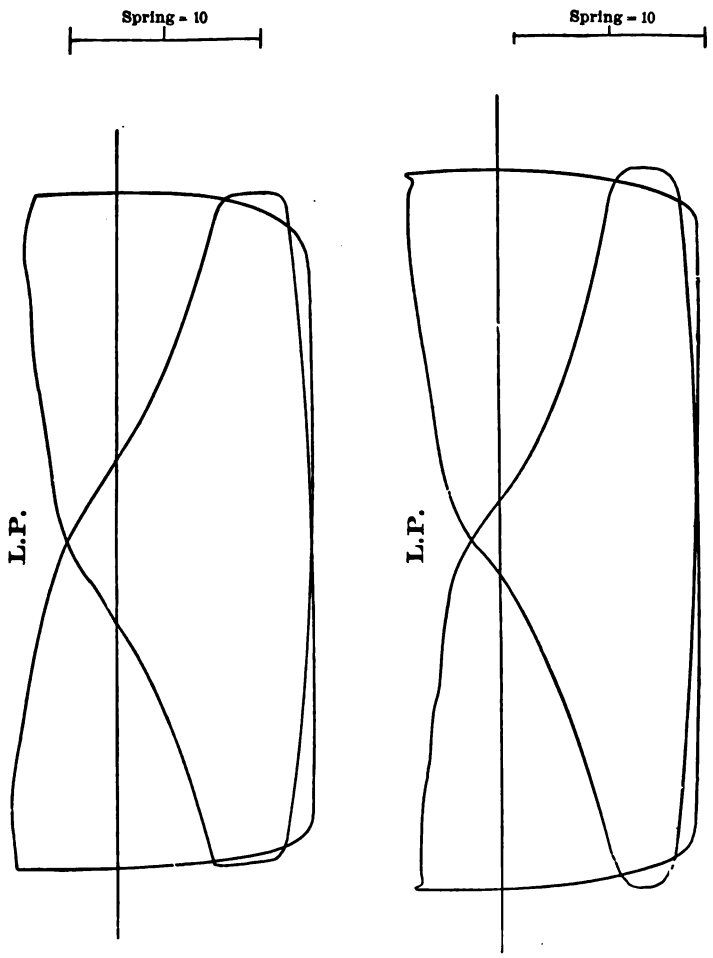
TRIPLE EXPANSION ENGINE
Cylinders 20," 32½"& 2-36"x 48" Stroke — 88 Revolutions.
FIG. 58.

It is no doubt in most cases economical to use steam-pipes of large capacity, to discard altogether ordinary cast elbows, employing only bent pipes of large radius, and to apply a receiver of ample capacity near the engine, all being thoroughly protected from radiation by the best known appliances.

The very rapid increase in the use of compound engines makes

it desirable, in discussing any valve-gear, to explain its manner of application to such engines. In tandem compounds of Buckeye construction, motion is imparted to the valves of the outboard cylinder by a very simple device, illustrated by Fig. 60.

The main valve of the first cylinder has a hollow stem on its



TRIPLE EXPANSION ENGINE
Cylinders 20", 32½" & 2 - 36" x 48" Stroke — 88 Revolutions.
FIG. 59.

outer end, which, projecting through the cylinder, is clamped to the hollow stem of the main valve of the second cylinder; the clamp connecting these stems is made in two parts, bolted together in such way as to produce universal adjustment.

One part of this double clamp carries a bearing in which

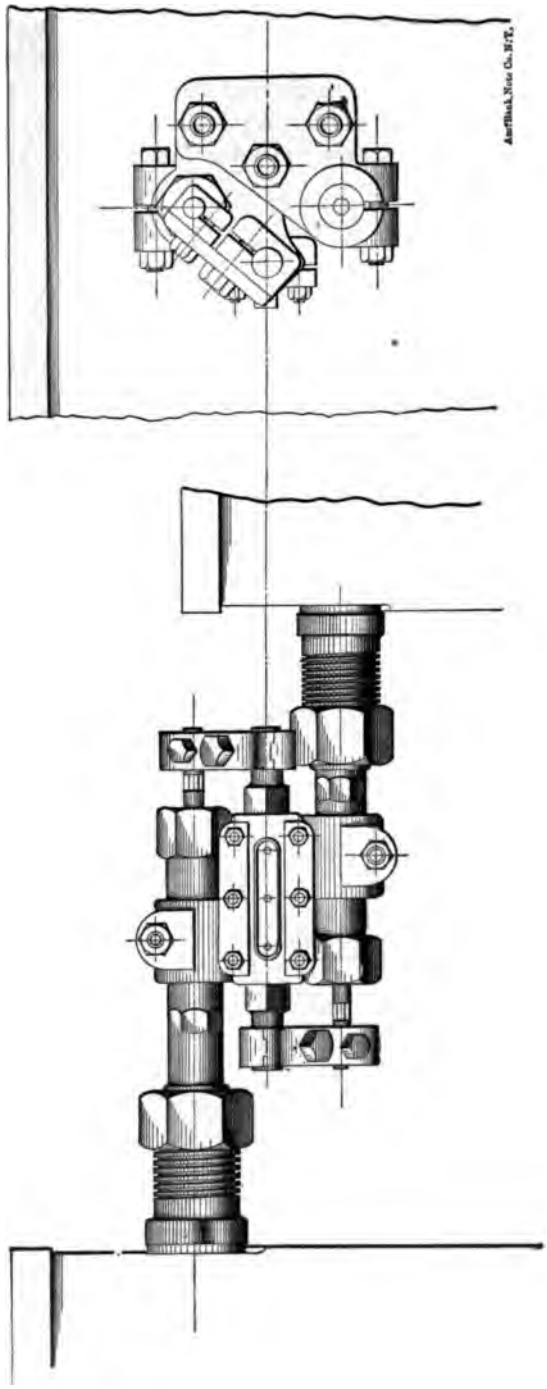


FIG. 60.

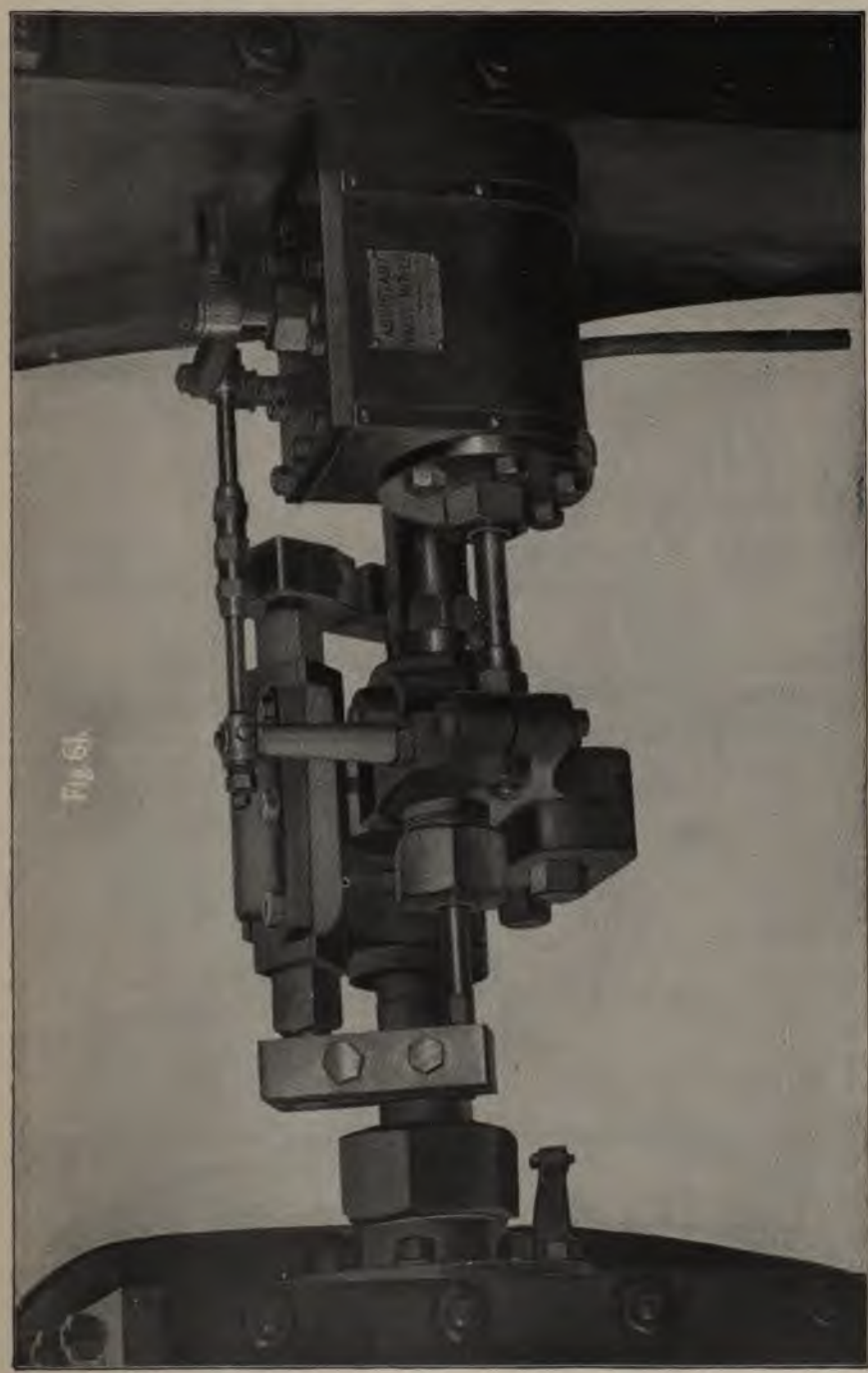


Fig. 61.

reciprocates a shaft, carrying at each end an adjustable arm; each arm is attached at its outer end to the cut-off valve stem of the cylinder adjacent to it. In this way one set of cut-off valves is

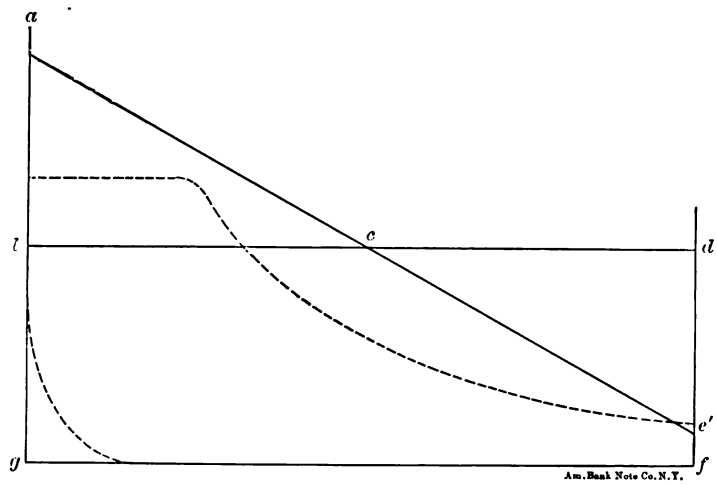


FIG. 62.

made a means of operating the other set without the use of complicated valve mechanism; and stuffing boxes surrounding all stems are easily accessible.

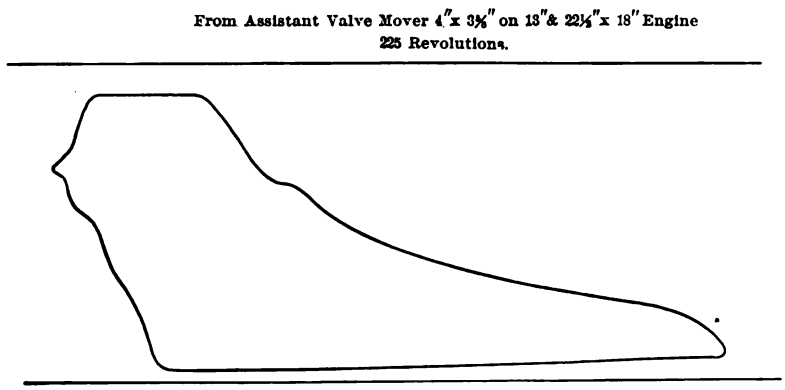


FIG. 63.

In Fig. 61 is shown the double clamp arrangement of Fig. 60, with the addition of another device called an "assistant valve-mover." This consists of a small steam-cylinder and piston taking hold of the valve-gear to assist the eccentric in the movement

of the valves. In compound engines of tandem construction and of large power the inertia of starting the valves and valve-gear at the beginning of each stroke is a considerable quantity, and

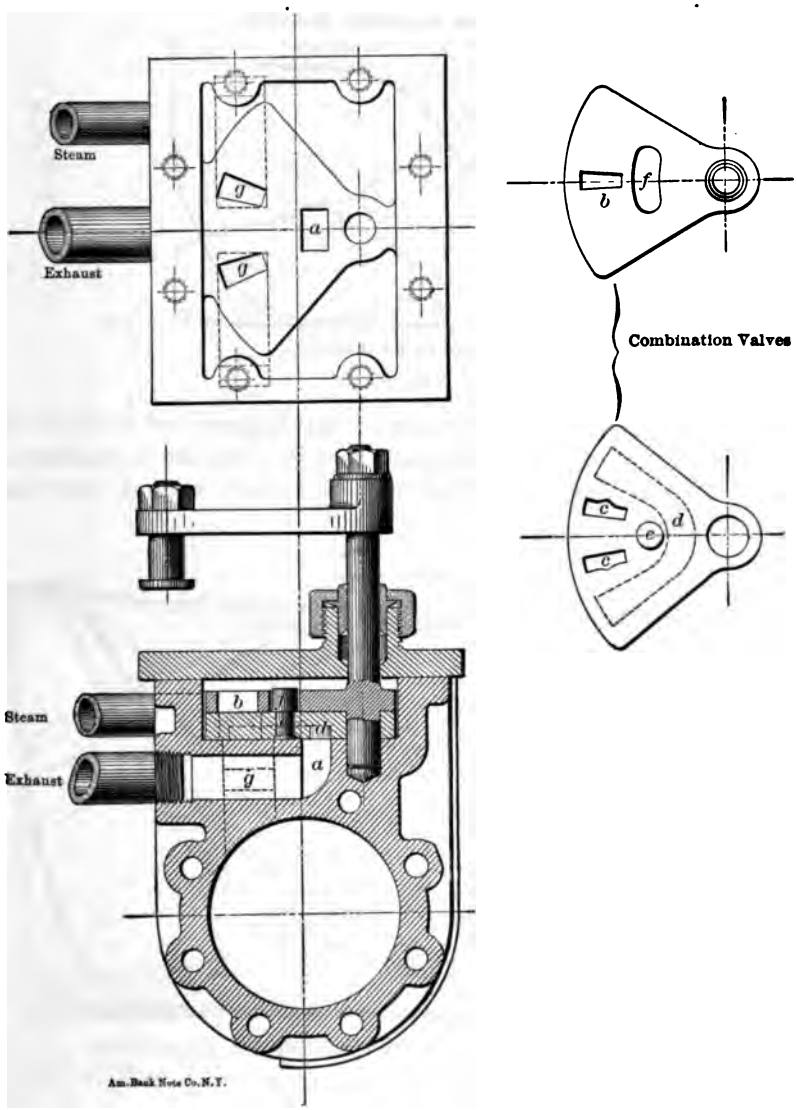
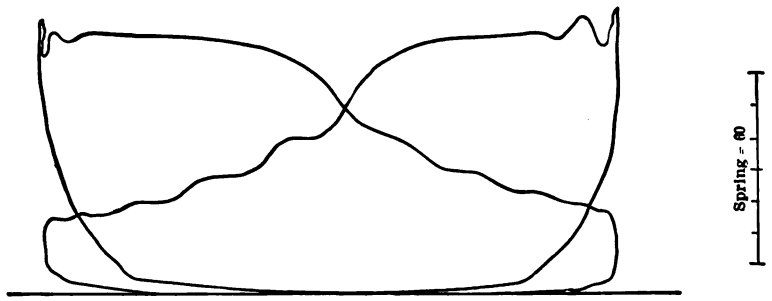


FIG. 64.

imposes a severe duty on the eccentric. This is the case whatever the kind of valves may be which are used in the engine.

The theory of the action of this cylinder may be illustrated as follows: Let *abcde* (Fig. 62) represent the well-known diagram of inertia of reciprocating parts (valves and valve-gear),

CARDS FROM BUCKEYE ENGINES.

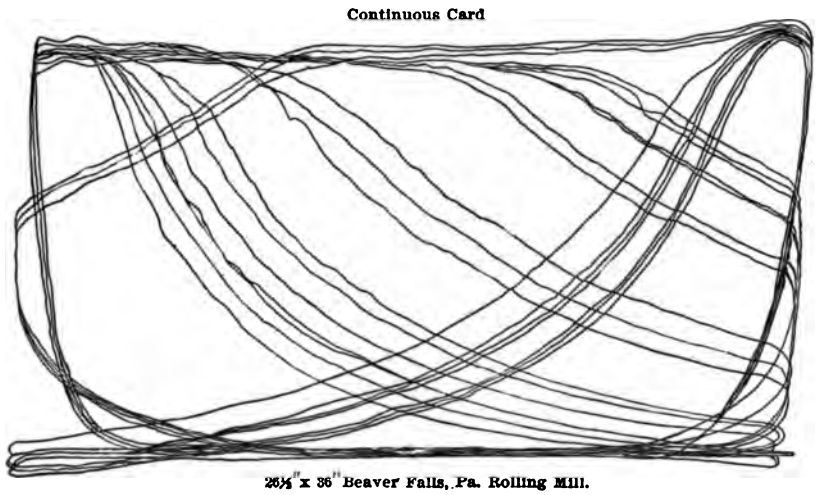


6½" x 10" Engine — 338 Revolutions.

FIG. 65.

and let the rectangle *bdfg* represent the diagram of friction of valves and gear on the assumption that this friction is constant, which it probably is not, but may be nearly enough for the present purpose.

Continuous Card



26½" x 36" Beaver Falls, Pa. Rolling Mill.

FIG. 66.

A combination of these diagrams leads to a complete diagram *acefg*.

Now an indicator card may be drawn, as shown by the dotted

lines, which approximates to this diagram, and the assistant cylinder is designed to so use the steam as to produce approximately this card. An actual card from such a cylinder is shown in Fig. 63. The cut-off is arranged to be constant and

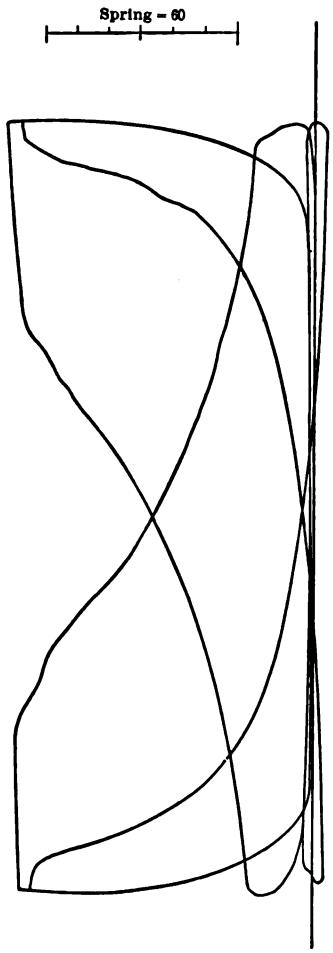


Fig. 67.

at about one-quarter stroke. The construction of valves and ports of this cylinder is shown by Fig. 64. It will be seen in Fig. 63 that the back-pressure line is such that the resultant pressure at and near the end of the stroke is a back pressure; this is thought to be best in the case of so high speed an engine as that from which the cards were taken. For the maxi-

mum inertia of the parts in such an engine is so much greater than the retarding friction, that (referring to Fig. 62) the line *ae* would run below the line *fg* at point *e*, making a back pressure at this point desirable. In slower running engines a slight change in adjustment does away with this back pressure.

DISCUSSION.

Mr. George I. Rockwood.—I would like to ask the gentleman one question, with reference to the statement made on page 188, just after the description of Fig. 57: "Most of the objections which have been urged against piston valves are overcome in this application. The travel of both main and cut-off valves being constant in extent on their seats, the wear is uniform."

I recall reading a lengthy description of the Buckeye Engine Company's exhibit at the World's Fair, written by the special commissioner of *The Engineer*, London, in the course of which this claim on the part of the Buckeye Engine Company, namely, the uniformity in extent of the travel of the cut-off valve on its seat, was controverted.

A diagram of the eccentric movements showed in what appeared to me at the time to be a plain enough demonstration, that this claim was incorrect. I would like to ask the gentleman to furnish proof of his statement.

Mr. J. F. Holloway.—While I will not undertake to describe in full the valve gear of the Buckeye Engine, I may say that the eccentricity of the eccentric which operates the valves in these engines is constant, and the travel of the valves in reference to each must also be the same. It is not like valves operated by a link motion, in which the throw of the valve varies as the link is moved up or down. The fact that steam can be cut off earlier or later in the stroke, is due to the fact that the eccentricity of the eccentric is revolved, more or less, around the shaft, the valves being moved by a rigid mechanism; in other words, it is the position of the valves with reference to the position of the piston, that controls the cutting off of the steam and the rate of the expansion.

It is true that while in some positions of the eccentric the cut-off valve will not make a full stroke with reference to any fixed object, it always does so in connection with the main valve; for while the travel of the cut-off valve may be shortened, the

movement of the main valve in an opposite direction will always be enough, when added to that of the cut-off valve, to make the total travel of the two valves equal to a constant maximum travel.

Mr. William Kent.—I would suggest that this rather complicated valve gear should be explained by a model. If the gentleman could find it convenient to make the Society a present of a model to illustrate this paper, it would be quite a good thing.

Mr. Mansfield.—That is a very apposite suggestion. We made a working model for the Exposition at Chicago which a great many of the members must have seen. I hope all did. It was a small engine made sectionally, horizontally, showing these facts that we are discussing completely. Now, since that time, there have been so many requests for that model from educational institutions and from societies of engine-men, etc., from all over the country, that we have finally determined to make some models, which shall be reproductions in fact of the engine—not models of the valve movement alone—with the cylinder and valves cut sectionally, horizontally, in order to show just what this paper states; and I believe I can say, without hesitation, that the Buckeye Engine Company will accept the invitation conveyed by the gentleman's suggestion, if it is agreeable to this Society.

Mr. Eckley B. Coxe.—As I understand the Buckeye valve gear, the cut-off valve moves a certain distance, just as the piston moves in the cylinder; the only difference is that in the case of the cylinder the latter stands still, whereas in the valve gear both parts are moving. The relative motion between the cut-off and the other valve is the same as the motion would be if the cylinder travelled on the piston. That which is made to vary is the time at which it passes a certain point. The cut-off valve starts from a certain point in the movable valve, goes to a certain other point and returns, so that, of course, it must always travel the same distance over the main valve.

Mr. F. H. Ball.—My understanding of this valve gear—(and I would like to be corrected if I am wrong)—is that the motion of the cut-off valve is introduced into the mechanism at a point where it makes no difference what the motion of the main valve is, or whether the main valve moves at all or not. This cut-off valve travels through a certain distance and a uniform distance relatively to the main valve, and therefore a certain distance by

the port in the main valve under all circumstances. The advancing of the eccentric simply changes the *time* when cut-off is effected, but does not change the speed at which it moves in cutting off, nor does it change the distance that it travels on the main valve. It would cut off exactly the same if the main valve should stand still, or if the travel of the main valve should be doubled.

Mr. W. S. Rogers.—I would like to ask Mr. Mansfield how, in his experience and practice, he finds the wear of this slide valve in service comparing with the piston valve? Does he not find leakage in the piston valve overcoming its value? I would like to have any information on this point he can give us.

Mr. James Hartness.—As I understand the matter, this cut-off eccentric has a fixed eccentricity to the main eccentric. Is that right?

Mr. Mansfield.—The cut-off eccentric has a fixed eccentricity to the main shaft, on which it is mounted. This is intended to be shown by Fig. 47. The interesting fact is, however, that

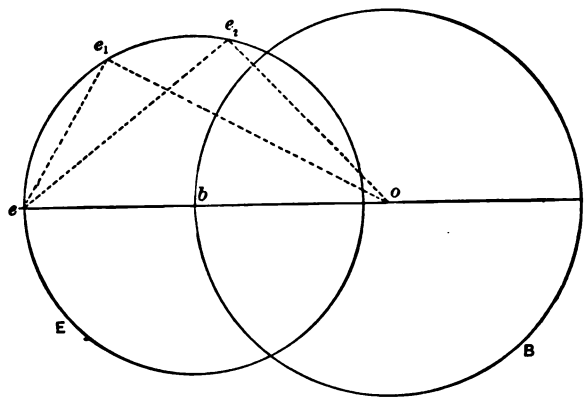


FIG. 68.

through the combination with the compound rocker-arm, the cut-off eccentric has *virtually* a fixed eccentricity to the main eccentric.

To illustrate this point, and at the same time to submit the proof asked for by Mr. Rockwood:

In Fig. 68 let *bo* represent the eccentricity of a main or fixed eccentric operating the main valve, and let *eb* represent the eccentricity of a governing eccentric operating the cut-off valve,

and so arranged that it is rotatively adjustable about the point *b* by means of the governor. Then, as will be shown, if connections to the valves be made directly from the points *e* and *b*, equivalent movements of the valves will be produced to those made by means of the rocker-arms and eccentrics of Fig. 47, and these movements will result in a uniform travel of the cut-off valve on the main valve, to whatever extent the point of cut-off may be varied.

In Fig. 69 let *b* and *e* again represent the centers of the eccentrics, rotative respectively about the centers *b* and *o* (*o* being the

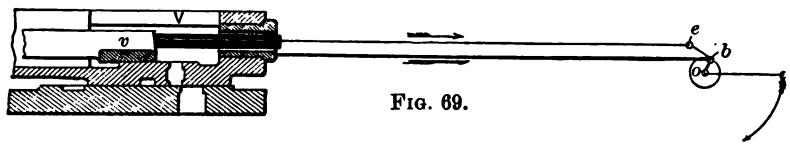


FIG. 69.

center of the shaft), and connected to their respective valves *V* and *v*.

First assume that the eccentric *b* stands still in the position shown, and that *e* is rotated about *b*. Clearly the travel of the cut-off valve on the main valve will be that due to its eccentric, and the case assumed is the common case of a valve driven by a fixed eccentric.

Suppose the eccentric *b* to be moved to *any other position* of its path and there fixed. Now rotate the eccentric *e* again about *b*; the action is the same as before—the cut-off valve travels the same distance on its seat and over the same path.

Now suppose *e*, *fixed relative to b*, and both to revolve about *o*. It will be seen that during each revolution of *b* about *o*, *e* makes a revolution about *b*, or relative to *b*; therefore the fact that *b* revolves, does not alter the extent of travel or location of path of valve *v* on its seat on valve *V*. Again, suppose *e* to be angularly adjustable about *b*, as by means of the governor, and let it be adjusted until the line *be* falls on *bo*. If *be* is equal in length to *bo*, then *e* coincides with *o*, and the *actual* travel of the cut-off valve becomes zero, yet its travel relative to the main valve is not changed, for *e* still makes a virtual revolution about *b* for each revolution of the fixed eccentric *b*.

To show that the result is exactly the same in the "Buckeye" combination, let us refer back to Fig. 47, in which *g* represents the main eccentric, and *h* the cut-off eccentric. The connection

of g to its valve is practically the same as in Fig. 69, although at an intermediate point of this connection the rocker-arm be is introduced, fixed at the point e and actuated at b by the main eccentric g.

Pivoted on the rocker-arm be, at about its middle point d, is the secondary rocker cdf, actuated at c by the eccentric h, of fixed eccentricity to the main shaft, and connected at f to the cut-off valve.

To prove that the cut-off valve travels equal distances, and on the same path, on the main valve, we have only to prove this relation between the points f and b, for the valves are rigidly connected to these points. For simplicity assume that the throw of the eccentrics is the same, also that $fd = bd = de = dc$. Next assume that h is adjusted about the shaft until it coincides with g. Then clearly the points b and c, Fig. 70, move with equal velocity in the same direction at the same time. Therefore, while b moves to b_1, c moves an equal distance to c_1, and d half as far to d_1. The point f remains in its original place, without movement. It is, however, as far to the left of b_1 as it was before to the right of b. It has, therefore, virtually moved *relative to b*, twice bf, equal to cc, equal to twice the eccentricity of the cut-off eccentric.

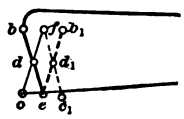


FIG. 70.

Suppose the point h, Fig. 47, to be adjusted to a position opposite to g, then the points b and c move with equal velocity in opposite directions at the same time, and we have the case of Fig. 71, in which c moves to c_1 while b moves to b_1, causing f to move a distance equal to the sum of the travel of both valves, but in its new position to be the same distance to the right of b_1 as it was before to the left of b, a distance equal to ce; therefore its path relative to b was exactly the same as before.

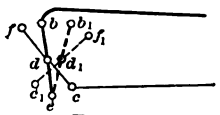


FIG. 71.

If h, Fig. 47, has any other angular position relative to g, then the points c and c_1 Figs. 70 and 71, representing extremes of movement, are not changed thereby, for the distance cc_1 is the distance due to the constant throw of the eccentric. From this it follows that $fb + f_1b_1$ must be constant, proving the constancy of the cut-off travel on the main valve.

The angular adjustment of the eccentric by the governor causes the point f to cross b at different points in the travel of

b, but always at the greatest velocity of *f*, since *f* crosses *b* when *c* is in mid-travel.

The cut-off therefore takes place at different positions of the main valve, but always equally sharp.

The cases of Figs. 70 and 71 are extreme cases not used in practice, since the angular adjustment of the cut-off eccentric never need be 180 degrees. Fig. 70 corresponds to a very late cut-off occurring when the main valve has not only closed but moved a distance equal to its lap after closure. Fig. 71 corresponds to a very early cut-off occurring when the main valve is in its mid-position and must move through its lap before opening.

It will be noted that the cut-off eccentric follows the main in Fig. 69, while it leads the main in Fig. 47. This is due in the latter case to the reversal of movement caused by the lever *fdc*, otherwise the combinations are practically identical.

It should be added in behalf of the "Special Commissioner" referred to by Mr. Rockwood, that the former has seen his error, and withdrawn his denial of the constancy of travel on its seat of the Buckeye cut-off valve.

In reply to Mr. Rogers, I have been experimenting in this piston-valve matter during the last two years, and at first had some question as to the result with reference to wear. But experience has answered the question satisfactorily. The constancy of travel of the valves seems to overcome all difficulties. The flat valves also wear well, due in great measure to the same cause. A constant travel seems to be a very important item in an engine valve.

DLXVIII.*

NOTES ON BELTING.

BY FRED. W. TAYLOR, PHILADELPHIA, PA.
(Member of the Society.)

INDEX TO NOTES ON BELTING, AND CONCLUSIONS TO BE DRAWN FROM THE PAPER.

In using belting so as to obtain the greatest economy and the most satisfactory results, the following rules should be observed:

Conclusions.	Oak tanned and fulled leather belts.	Other types of leather belts, and 6 to 7 ply rubber belts.	Refer to paragraph No.
No. 1. A double belt, having an arc of contact of 180°, will give an effective pull on the face of a pulley per inch of width of belt of.....	35 lbs.	30 lbs.	(81), (82), (84), (85); (22) & (26) to (80). Driving power.
Or a different form of same rule: 1 The number of square feet of double belt passing around a pulley per minute required to transmit one horse-power is.....	80 sq. ft.	90 sq. ft.	
Or: 1. The number of lineal feet of double belting 1 inch wide passing around a pulley per minute required to transmit one horse-power is.....	950 ft.	1,100 ft.	

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

Conclusions.	Oak tanned and fulled leather belts.	Other types of leather belts, and 6 to 7 ply rubber belts.	Refer to paragraph No.
No. Or: 1. A double belt 6 inches wide, running 4,000 to 4,500 feet per minute, will transmit	30 H.P.	25 H.P.	

No.	Conclusions.	Refer to Paragraph No.
2.	The belt speed for maximum economy should be from 4,000 to 4,500 feet per minute.	(87), (88), (83) to (87). Speed.
3.	The best distance from centre to centre of shafts is from 20 to 25 feet.	(111). Centre to centre.
4.	The faces of pulleys should be about 25% wider than their belts.	(112). Face of pulleys.
5.	When it is necessary to run night and day through the week without stopping, each important belt should be supplied with an idler pulley which can be tightened upon it while running, in case of slip.	(108). Idler pulleys.
6.	Idler pulleys work most satisfactorily when located on the slack slide of the belt about one-quarter way from the driving pulley.	(109). Idler location.
7.	Belts are more durable and work more satisfactorily made narrow and thick, rather than wide and thin.	(95) to (103). Thickness.
8.	It is advisable to use double belts on pulleys 12 inches diameter, or larger. It is advisable to use triple belts on pulleys 20 inches diameter, or larger. It is advisable to use quadruple belts on pulleys 30 inches diameter, or larger.	(103) and (95) to (102). Minimum diameter of pulley for given thickness.
9.	As belts increase in width they should also be made thicker.	(101) and (102). Ratio of thickness to width.
10.	The ends of the belt should be fastened together by splicing and cementing, instead of lacing, wiring, or using hooks or clamps of any kind.	(104) to (107), (8). Splice.

No.	Conclusions.	Refer to Paragraph No.
11.	V-splice should be used on triple and quadruple belts and when idlers are used. Stepped splice, coated with rubber and vulcanized in place, best for rubber belts.	(104) to (107). Splice.
12.	When practicable, belts should be tightened by moving one pulley away from the other. Countershafts should be mounted on frames, and raised in tightening vertical and diagonal belts.	(110) and (9). Tightening, best method.
13.	Belt clamps having spring balances between the two pairs of clamps should be used for weighing the tension of the belt accurately each time it is tightened.	(122) and (10), (11), (74), and (68) to (73). Spring balance belt clamps.
14.	When it is impracticable to accurately weigh the tension of a belt in tightening it, it is safe to shorten a double belt one-half inch for every 10 feet of length, if it requires tightening, when working, according to the ordinary rules, under a total load of 111 lbs. per inch of width and giving an effective pull of 65 lbs. per inch of width. If it works under a total load of 54 lbs. and effective pull of 26 lbs., it is safe to shorten the belt one inch for every ten feet of length, when it requires tightening.	(122), (59), and (58). Tightening. Amount to cut out.
15.	Belts should be cleaned and greased every five to six months.	(120) and (9). Greasing.
16.	Belts should be tightened and repaired and cared for out of working hours by one man as far as practicable, careful inspection as to their condition being made at regular intervals.	(124), (123) and (15), (10). Care and inspection.
17.	Double leather belts will last well when repeatedly tightened under a strain (when at rest) of 71 lbs. per inch of width, or 240 lbs. per square inch section. They will not maintain this tension for any length of time, however.	(121), (46), (40). Tension to which belts can be tightened.

No.	Conclusions.	Refer to Paragraph No.
18.	If double leather belts are tightened while at rest to 71 lbs. per inch of width, and subjected to an additional working load of 65 lbs. per inch of width, their tension will fall in two and one-half months so as to be, while at rest, 33 lbs. per inch width, or 106 lbs. per square inch section; Their average tension during these two and one-half months being 46 lbs. per inch width, or 150 lbs. per square inch section; Their average total load during these two and one-half months being 111 lbs. per inch width, or 358 lbs. per square inch section. These are the conditions under which belts work when tightened according to the ordinary rules.	(52), (53), (48) to (55), (73), (77) to (80). Fall in tension. Average tension. Average total load under ordinary rule.
19.	When double belts are first tightened to 71 lbs. per inch of width, and then subjected to an additional working load of 26 lbs. per inch of width, their tension falls in twenty-two months to 21 lbs. per inch width, or 68 lbs. per square inch section; The average tension during this time being 28 lbs. per inch width, 90 lbs. per square inch section; and the Average total load being 54 lbs. per inch width, 174 lbs. per square inch section. These conditions are rather less severe on belting than the rules which we recommend as most economical.	(48) to (55), (73), (77) to (80). Fall in tension. Average tension. Average total load under more economical load rule.
20.	The most economical average total load for double belting is 65 to 73 lbs. per inch of width—*i.e.*, 200 to 225 lbs. per square inch section. This corresponds to an effective pulling power of 30 lbs. per inch of width, and with rules No. 1, given above.	(80) and (84) to (86). (72) to (75), (77), (78), (79). Most economical total load.

No.	Conclusions.	Refer to Paragraph No.
21.	Double leather belts, when treated with great care and run night and day at moderate speed, should last for seven years, when under average total load of 111 lbs. per inch width, or 358 lbs. per square inch section. They should last for 18 years when under average total load of 54 lbs. per inch width, 174 lbs. per square inch section. A total load of 111 lbs. per inch width corresponds to a pulling power of 65 lbs. per inch width. A total load of 54 lbs. per inch width corresponds to a pulling power of 26 lbs. per inch width.	(40). See footnotes. (63), (64), (54), and (50). Life of belting.
22.	The total life of belting, cost of maintenance and repairs and the interruptions to manufacture caused by belts, are dependent upon the total load to which the belts are subjected, more than upon any other condition. The other conditions chiefly affecting the durability of belting being : 1st. Whether spliced, or fastened with lacing or belt hooks. 2d. Whether they are properly greased and kept clean and free from machine oil. 3d. The speed at which they run.	(73), (63) to (74); also (104), (121), (86), (71). Life of belting dependent upon total load and other elements.
23.	The speed at which belting runs has comparatively little effect on its life until it passes 2,500 to 3,000 feet per minute.	(86) to (89), and (71). Life not affected by speed.
24.	In a machine shop, the average cost in dollars per double belt per year of service, including original cost and cost of maintenance and repairs, proved to be $6.72 for such belts as were used under the ordinary rules (111 lbs. total load per inch of width, giving an effective pull of 65 lbs. per inch of width), and the cost was found to be less than $5.70 for belts under total load of 54 lbs., effective pull of 26 lbs., per inch of width.	(41), (32) to (37), (54) and (50). Total cost per year.

No.	Conclusions.	Refer to Paragraph No.
25.	The cost of all labor and materials used in the maintenance and repairs of double belts, added to the cost of renewals as they give out, through a term of years, will amount on an average per year to 37% of the original cost of the belts, if the belts are tightened according to the ordinary rules, and work under a total load of 111 lbs. per inch of width, giving an effective pull of 65 lbs. per inch of width. If, however, the belts are working according to a more economical rule, under a total load of 54 lbs. per inch of width, and an effective pull of 26 lbs. per inch of width, the annual cost of maintenance, repairs, and renewals amounts to not more than 14% of the first cost, probably less.	(125) and (42), and (32) to (41), (54), (50). Ratio of total cost per year to first cost.
26.	The cost of maintenance and repairs of double leather belts amounts through a total life of 6.7 years (running night and day) to one and one-half times the first cost of the belts, when belts are working, according to the ordinary rules, under a total load of 111 lbs. per inch of width. and are giving an effective pull of 65 lbs. per inch of width. When, however, the belts are working according to a more economical rule, under a total load of 54 lbs. per inch of width, and give an effective pull of 26 lbs. per inch of width, the cost of maintenance and repairs, through a term of 8.8 years, amounts to only 30.4% of their first cost.	(66) and (65), (34), (32) and (33), (54) and (50). Cost of maintenance and repairs.
27.	The total stretch of leather belting certainly exceeds 6% of the original length.	(43) and (44), (89). Total stretch.

No.	Conclusions.	Refer to Paragraph No.
28.	The stretch during the first six months of the life of belts is 36% of their entire stretch, in the case of double belts working, according to ordinary rules, under a total load of 111 lbs. per inch of width, and giving an effective pull of 65 lbs. per inch of width. If, however, the belts are working according to a more economical rule, under a total load of 54 lbs., giving an effective pull of 26 lbs., the stretch during the first six months is 15% of entire stretch.	(60), (92), (89). Stretch during first six months.
29.	A double belt will stretch $\frac{47}{100}$ of 1% of its length before requiring to be tightened, when subject, according to ordinary rules, to a total load of 111 lbs. per inch of width, and giving an effective pull of 65 lbs. per inch of width. A double belt with total load of 54 lbs., effective pull of 26 lbs., stretches $\frac{81}{100}$ of 1% before requiring tightening.	(58), (122), (73), (54), (50). Stretch between tightenings.
30.	The most important consideration in making up tables and rules for the use and care of belting is how to secure the minimum of interruptions to manufacture from this source.	(4), (2), (3), (61), (62), (84), (85). Interruptions to manufacture.
31.	The average double belt, when running night and day in a machine shop, will cause at least 26 interruptions to manufacture during its life, or 5 interruptions per year, if the belts work, according to ordinary rules, under a total load of 111 lbs. per inch of width, and give an effective pull of 65 lbs. per inch of width. If, however, they work according to a more economical rule, under a total load of 54 lbs., giving an effective pull of 26 lbs., then the interruptions to manufacture will not average oftener for each belt than one in sixteen months.	(67), (61), (62), (54) and (50). Interruptions to manufacture.

No.	Conclusions.	Refer to Paragraph No.
32.	In a carefully conducted experiment between different types of leather belts, made in a machine shop, lasting 9 years, between 40 and 50 belts being used, the oak tanned and fulled belts showed themselves to be superior in all respects except the coefficient of friction to either the oak tanned not fulled, the semi-raw-hide, or raw-hide with tanned face. They lasted longer, caused fewer interruptions to manufacture, did not stretch unevenly, stretched less, cost less per year of service, required tightening only half as often, and gave less trouble when first started than their competitors.	(91) to (95), (16), (17). Also in general (4) to (62). Oak tanned and fulled better than other types of belts.
33.	Belts of any width can be successfully shifted backward and forward on tight and loose pulleys. Belts running between 5,000 and 6,000 feet per minute, and driving 300 H.P., are now being daily shifted on tight and loose pulleys, to throw lines of shafting in and out of use.	(113) and (117). Large shifting belts.
34.	Shifting belts are much to be preferred to cut-off couplings, or friction clutch pulleys, for throwing heavy lines of shafting in and out of use.	(117) and (113). Shifting belts preferred to friction clutch pulleys or cut-off couplings.
35.	The best form of belt shifter for wide belts is a pair of rollers twice the width of belt, either of which can be pressed onto the flat surface of the belt on its slack side close to the driven pulley, the axis of the roller making an angle of 75° with the centre line of the belt.	(115) to (118). Belt shifter for wide belts.
36.	Loose pulleys of the McCaffrey type would appear to be more durable than others.	(114). Good loose pulley.

1. The problems in belting, while apparently at first sight so simple, are in reality so complicated, and so many considera-

tions must be entertained in their solution, that most of us find ourselves continually forming new conclusions on the subject and revising old ones. The notes contained in this paper are offered not as a strictly logical and systematic treatise, but with a wish to present for discussion some rather disconnected conclusions arrived at from practical rather than theoretical considerations.

2. It would appear from a study of the best experiments and articles on belting, from which the rules and tables in common use have been deduced, that the chief consideration has been how to get the maximum of work from belting; while, in making up belting tables, the two most important considerations—how to secure the minimum of interruptions to manufacture, and the maximum of durability—have not hitherto received due attention. The one consideration which should have more weight than all others in making up tables and rules for the use and care of belting is how to secure the least possible interruption to manufacture from this source.

3. It is the writer's judgment that belts should be made heavier and run more slowly than theory and accepted rules would indicate, not only for the sake of reducing the belt bill in the long run, but even more to avoid the frequent interruptions to manufacture. In figuring the total expense of belting, and the manufacturing cost chargeable to this account, I think that most careful observers soon come to the conclusion that by far the largest item in this account is the time lost on the machines while belts are being relaced and repaired. This is certainly the case even where the process of manufacture is such that any one machine can be stopped without affecting the running of its neighbors, but far more so in those establishments where the running of a series of machines is dependent one upon another, and the stoppage of one machine involves delays on others.

A NINE YEARS' EXPERIMENT ON BELTING.*

4. While working as foreman of a machine shop, the tools of

* This experiment was made in the machine shop of the Midvale Steel Co., of Philadelphia, beginning in 1884 and lasting until 1893. During the first six years the experiment was under the direction of the writer, while he successively occupied the position of foreman of machine shop, master mechanic, head of the drafting room, and chief engineer, after which time it was carefully conducted by his successor.

which were frequently driven to their maximum capacity, the writer became convinced that the belts, which were laced according to the ordinary rules, were a great source of loss to the company—not so much from the cost of the belting and the labor of lacing as from the incidental delays to the machines, and the diminished output of the shop resulting therefrom. This was particularly emphasized when piece-work rates were established, requiring the machines to be driven hard and continuously. The belting was then shown to be by far the largest source of trouble in the shop.

5. In looking into the experiments that had been made and the articles that had been written on belting up to this time, it was found that all carefully conducted and recorded experiments were of short duration, their chief object being the determination of the coefficient of friction. But of equal importance in formulating rules for belting is the knowledge of what tension can be surely maintained through a term of months, or what elements chiefly affect the durability of belting; yet these considerations appear to have been rather neglected by experimenters. Very little information could be obtained either as to the cost of maintenance of belts, or in regard to the interruptions to manufacture from belting, when used under known and uniform conditions as to tension and general treatment.

6. Believing that reliable information on these points could only be obtained by means of an experiment lasting through a term of years, the writer was fortunate in having the opportunity of making a trial of this sort; and although at the end of nine years the experiment is not finished, still it has proceeded far enough to enable him to form conclusions which are interesting and, it is believed, useful.

7. In 1883 and 1884 the writer superintended the design and erection of a new machine shop, in which many of the machines from an old shop were used.

8. As a result of experience in the old shop, the tight and loose pulleys on the countershafts were made much larger in diameter and of wider face, so that the belt power from main line to countershafting was made about two and one-half times as great as formerly. All belts were made endless by splicing, glueing, and pegging, instead of lacing or hooking, and *double belts* were used throughout the shop.

It was found impracticable in most cases to alter the driving

pulleys from the countershaft to the machine, since generally cone pulleys were used.

9. In all cases the countershafts were mounted on independent frames, which could be raised and lowered in tightening the belts by the interposition of wooden packing pieces of varying thickness between the frames and the supporting stringers overhead. For this purpose standard packing pieces, varying by eighths of an inch in thickness, were always kept in the tool room. With this method of tightening it was seldom necessary to resplice a belt, since six to ten inches of stretch could be taken up in the belt, by gradually raising the countershaft, before resplicing became necessary.

10. Belt clamps were used having spring balances between the two pairs of clamps, so that the exact tension to which the belt was subjected was accurately weighed when the belt was first put on, and each time it was tightened.

11. Experience soon demonstrated about the length of time that each belt would run without requiring to be tightened, and at approximately regular periods the spring-balance belt clamps were put on to each belt and the tension of same weighed, and the countershaft raised just enough to maintain the belt at its proper tension. For this reason, it was a matter of very rare occurrence that a belt slipped during working hours. And as the belts were generally tightened on Sundays (the shop working night and day), the minimum of delay was caused on the machines from this source.

12. In case a belt slipped while the machine was in use, the man running it was never allowed to take it up. It was reported to the foreman, and one man—as far as possible—took care of all the belts. He received his orders from and made his returns in writing to the machine-shop clerk, concerning time and materials used, tension of the belts, kind and condition of the splices, and nature of repairs made.

13. The tension under which each belt was spliced was carefully figured so as to place it under an initial strain—while the belt was at rest immediately after tightening—of 71 lbs. per inch of width of double belts.

This is equivalent, in the case of

Oak tanned and fulled belts,	to 192	lbs.	per	square	inch	section.
Oak tanned, not fulled belts,	" 229	"	"	"	"	"
Semi-raw-hide belts,	" 253	"	"	"	"	"
Raw-hide belts,	" 284	"	"	"	"	"

14. At intervals of about three months for the first two years of the test, and after this time at intervals of about five months, each belt was scraped clean, and greased with the kind of dubbing recommended by the maker of the belt.

15. An accurate account was kept of the original cost of each belt, and every item of expenditure, both for labor and materials used in the maintenance and care of same; also the exact stretch of each belt was recorded, and its method of treatment throughout.

16. Having determined upon the above method of treating the belts before building the shop, the writer concluded that this would afford an excellent opportunity for testing several types of leather belting. He therefore selected what he believed to be the four leading types of leather belts, choosing the firm with the best reputation for each type, and used the types in about equal numbers through the shop. As the shop contained a good many machines of the same kind, which were run under the same conditions, it was possible to subject the competing belts to practically the same conditions. The makers were all informed of the nature of the test which was to be made, and were given plenty of time in each case to make the best belts of their types, so that none but first-class belts of their kinds were used.

17. The following were the types of belts selected:

1. Oak tanned and fulled leather.
2. Oak tanned but not fulled.
3. Semi-raw-hide.
4. Raw-hide with surface tanned.

18. In considering the results of this test, it should be borne in mind that the belts called "shifting" are those running from the main line of shafting to tight and loose pulleys on the countershafts, and that these belts were used so as to have about two and one half times as great transmitting power as the ordinary belting rules would demand; while the "cone" belts, extending from the countershaft to the machine, are used according to the ordinary rules for belting.

19. Below will be found a summary of the results of the test obtained from a study of the records. In considering these averages, it should be borne in mind that the test was begun in 1884, since which time (for about nine years) the "shifting" belts have been run on an average twenty hours per day, while

the "cone" belts have not received quite such steady use, since the machines were stopped while work was put on and taken off, and they have been occasionally idle owing to a lack of work. Still, they have certainly been used much more steadily than those in the average machine shop.

20. It is to be regretted that the records, giving the exact details of the cost of repairs, tightening, and stretch, etc., of each belt, are so voluminous as to prevent their publication. They would, however, fill nearly one-half of a volume of the proceedings of the Society.

The records of many of the individual belts are erratic in many respects, so that it is only by a series of averages that we apparently can arrive at any reliable conclusions.

21. The writer in this paper frequently uses the expression, "the ordinary rules of belting." A glance at *The Use of Belting*, a book in which Mr. John H. Cooper has carefully compiled all of the belting rules established by the best authorities abroad and in this country, shows us that these rules present a wide range of difference, the ratio of divergence between the lightest and heaviest rules being as 1 to 4. By the expression, "the ordinary rules of belting," we refer to a mean of these various rules, which gives a double belt a pulling power of 65 lbs. per inch of width, and places the most economical belt speed at between 5,000 and 6,000 feet per minute.

In reading articles on belting, we note that there seems to be no uniformity in the use of terms to express the tension under which belts are tightened, and the loads to which they are subjected while running. We therefore think it advisable to define the meaning which we attach to certain terms.

When pulleys upon which belts are tightened are at rest, both strands of the belt (the upper and lower) are under the same stress per inch of width. By "tension," "initial tension," or "tension while at rest," we mean the stress per inch of width, or square inch of section, to which one of the strands of the belt is tightened, when at rest. After the belts are in motion and transmitting power, the stress on the slack side, or strand, of the belt becomes less, while that on the tight side—or the side which does the pulling—becomes greater, than when the belt was at rest. By the term "total load," we mean the total stress per inch of width, or square inch of section, on the tight side of belt while in motion.

The difference between the stress on the tight side of the belt and its slack side, while in motion, represents the effective force or pull which is transmitted from one pulley to another. By the terms "working load," "net working load," or "effective pull," we mean the difference in the tension of the tight and slack sides of the belt per inch of width, or square inch section, while in motion, or the net effective force that is transmitted from one pulley to another per inch of width or square inch of section.

22. In examining the summary of the following table, please bear in mind that shifting belts were so used as to be subject to a net working load only $\frac{4}{10}$ as great as that of the cone belts (see 10 and 18). Each style of belt was, however, tightened repeatedly to the same tension, 71 lbs. per inch of width. All belts were double leather.

The cone belts were *supposed* to give an effective pull on the pulley of 65 lbs. per inch of width, and the shifting belts 26 lbs.

23. Originally about fifty belts were started in this experiment, but only forty-two were found available for making up the averages given beneath.

24. The shifting belts vary in dimensions from 39 feet 7 inches long, 3.5 inches wide, .25 inch thick, to 51 feet 5 inches long, 6.5 inches wide, .37 inch thick. Cone belts vary in dimensions from 24 feet 7 inches long, 2 inches wide, .25 inch thick, to 31 feet 10 inches long, 4 inches wide, .37 inch thick.

25. The shifting belts vary in speed from 565 feet per minute to 1,570 feet per minute. Cone belts vary in speed from 225 feet per minute to 1,340 feet per minute.

SUMMARY OF RESULTS OF EXPERIMENT IN LEATHER BELTING LASTING NINE YEARS.

Item No.		Shifting Belts.					Cone Belts.						
		Oak tanned and fulled.	Oak tanned, not fulled.	Semi-raw-hide.	Raw-hide.	Average of all shifting belts.	Average of all cone belts.	Oak tanned and fulled.	Oak tanned, not fulled.	Semi-raw-hide.	Raw-hide.	Item No.	
26	Average length of belts in feet	49	Too few of this style of belts were used to make up a proper average.	50	54	50	29	30	28	29	30	26	Length.
27	Average width of belts in inches	5		5	4.50	4.84	3.8	3.82	3.31	4.5	3.75	27	Width.
28	Average thickness of belts in hundredths of an inch	.37		.28	.25	.31	.31	.37	.31	.28	.25	28	Thickness.
29	Average speed of belts in feet per minute	.....		.....	.....	1,140	580	.....	.....	.....	.....	29	Speed.
30	Percentage of belts still in use after 9 yrs. use (20 hrs. per day) *	All *		All	80	93.33*	27*	57	None	25	20	30	Still in use.
31	Percentage of belts worn out during 9 years	None		None	20	6.66	73	43	All	75	80	31	Worn out.
32	Average original cost per belt in dollars	36.44		46.17	43.53	40.16	18.04	16.34	13.53	22.42	21.88	32	First cost.
33	Average cost per belt of cleaning, greasing, tightening, and repairing, in dollars	9.80		14.66	14.86	12.23	27	24.27	19.18	31.18	36.05	33	Maintenance.
34	Average ratio that care and maintenance bears to first cost, %	0.27		0.31	0.34	0.30.4	1.50	1.49	1.42	1.39	1.65	34	Ratio.
35	Average cost per belt of cleaning and greasing, in dollars	4.41		4.50	3.74	4.24	2.56	2.68	2.64	2.41	2.40	35	Greasing.
36	Average cost per belt of tightening, in dollars	1.90		2.31	2.69	2.24	4.64	4.31	3.51	5.23	5.83	36	Tightening.
37	Average cost per belt of repairing, in dollars	3.37		7.81	8.42	5.75	19.88	17.27	13.03	23.53	27.90	37	Repairs.
38	Average cost of belt grease used in dressing belts			10 cents per belt per year.								38	Belt grease.
39	Average cost of belt glue used in splicing and repairing			13 cents per belt per year.								39	Glue.
40	Average number of years belts lasted †	9		9	8½	8.8*	6.7†	7.2	5.5†	8.5†	5.8†	40	Life.
41	Average cost in dollars per belt per year of service, including original cost and cost of maintenance and repairs	5.13		6.76	6.87	5.70	6.72	5.64	5.95	6.30	9.98	41	Cost per year.
42	Ratio that total cost per year of life of belt, as given in Item 41, bears to first cost of belt	14%		14.6%	15.8%	14.2%	37.3%	34.2%	44%	28%	45.4%	42	
43	Total stretch of belts in inches as entered in record ‡	11.9		16.9	17.3	15‡	21.4‡	16.7	21	21.4	28	43	Stretch.
44	Ratio of total stretch to total length of belt, %‡	.02		.02.8	.02.7	.02.3‡	.06‡	.04.6	.06.2	.06.6	.07.8	44	Stretch.
45	Average number of times each belt required tightening ‡	2.66		3	3.6	3‡	14.5‡	12	12	16	20	45	Tightened.
46	Tension in lbs. per inch of width to which belts were strained each time that they required tightening	71		71	71	71	71	71	71	71	71	46	Initial tension.
47	Tension in lbs. per square inch of section of belt to which belts were strained each time that they required tightening	192		253	284	239	239	192	229	253	284	47	When at rest.
48	Average strain in lbs. per inch of width of belt to which the tension of belts had fallen when they required tightening	22.7		21.5	19.3	21.1	32.9	37.6	30	29.6	34.3	48	Fall in tension.

Item No.		Shifting Belts.					Cone Belts.						
		Oak tanned and fulled.	Oak tanned, not fulled.	Semi-raw-hide.	Raw-hide.	Average of all shifting belts.	Average of all cone belts.	Oak tanned and fulled.	Oak tanned, not fulled.	Semi-raw-hide.	Raw-hide.	Item No.	
49	Average strain in lbs. per square inch of section of belt to which tension of belts had fallen when they required tightening	61	Too few of this style of belts were used to make up a proper average.	77	77	68	106	102	97	106	133	49	
50	Average net (estimated) working load in lbs. per inch of width of belt. This is the effective pull per inch of width which the belts were supposed to give on the rim of the pulley..	26		26	26	26	65	65	65	65	65	50	Working load.
51	Av. net (est.) working load in lbs. per sq. in. section of belt ..	70		93	104	84	210	175	210	232	260	51	
52	(Estimated) average tension in lbs. per inch of width of belt exclusive of working load.....	32		26.5	28.5	28	46	49	44	43	47	52	Average tension.
53	(Estimated) average tension in lbs. per square inch section of belt	86.4		95	114	90	151	133	142	154	188	53	When at rest.
54	Average total (estimated) load (including working load and tension) in lbs. per inch of width.....	58		52.5	54.5	54	111	114	109	108	102	54	Total load.
55	Average total load in lbs. per square inch section of belt (estimated.	157		188	218	174	358	308	351	385	408	55	
56	Average number of months from the time that belts were tightened to a tension of 71 lbs. per inch of width until they required re-tightening.	32		12.9	21	22	2.52	4.41	1.30	1.66	2.71	56	Time between tightening.
57	Average number of inches that each belt stretched from the time it was tightened until it next required tightening...	4.48		5.31	5.56	5.11	1.55	1.52	1.82	1.43	1.45	57	Stretch.
58	Ratio of stretch between tightenings to total length of belt.	0.73%		0.83%	0.87%	0.81%	0.47%	0.41%	0.61%	0.40%	0.47%	58	Stretch.
59	Average stretch per belt in inches that takes place in first six months	None recorded		3.2	3.3	2.2	7.8	3.08	7.8	6.7	14	59	Stretch.
60	Percentage of the total stretch of belt that takes place during the first 6 months ‡	.....		19	19	14.6‡	36‡	18.4	37	31	50.6	60	Stretch.
61	Average number of times each belt was tightened or repaired..	5.5		9.27	10	6.6	32	29	26	37	40	61	Interruption to manufacture.
62	Average number of times per year of life of belt that it was either tightened or repaired.....	0.6 times		1.08	1.18	0.75	5 times†	4	4.7†	4.4†	7†	62	

* An examination of the shifting belts still in use shows them to be practically about as good as new, while the cone belts still in use are nearly worn out.

† Among the cone belts, two oak tanned not fulled, five semi-raw-hide, and one raw-hide belt stretched unevenly, and were returned to their makers a few months after being put into use, and these were not taken into consideration in the averages of Item 40, although they were considered in averaging Item 30. None of the oak tanned and fulled belts stretched unevenly.

‡ I do not consider the data regarding total stretch of cone belts as at all reliable. For the first few years the records of stretch were accurately kept, but when the belts began to require repairs, this portion of the record was carelessly kept, since frequently the belts (see Item 45) were tightened at the same time that they were repaired; the total stretch of the cone belts (see Item 43) certainly exceeded that given in the table. The data regarding the stretch of belts given in Items 57, 58, and 59 is reliable, as only instances concerning which there was no question were considered in making up the averages.

63. In considering the above table, the most interesting and important fact noticeable is the superiority of the shifting to the cone belts in every respect except first cost (32), and this superiority is even much greater than the figures would indicate, since, generally speaking, the cone belts which are still in use are nearly worn out, having reached a point at which it is doubtful whether it pays to repair them, while the shifting belts are, to all appearances, in almost as good condition as when they first went into use, and should last twice as long as they have already.

64. I think it would be safe to say that the life of the shifting belts will be three times that of the cone (see 30 and 40, with foot-notes), and already the total cost of the shifting belts per year of service is less than that of the cone. (See 41.)

65. A comparison of the first cost of the belts (32) is scarcely fair to the shifting belts, since, owing to the location of the machines, the cone belts are only two-thirds as long as their competitors (26).

66. It is interesting to note that after 8.8 years of life the total cost of maintenance and repairs of the shifting belts (33 and 34) amounts to only 30.4% of the original cost, while with the cone belts the maintenance and repairs through a life of 6.7 years amounts to one and one-half times the first cost.

67. In the writer's judgment, by far the greatest point of advantage of the shifting belts lies in the fact that the interruptions to manufacture are nearly seven times as frequent with the cone as with the shifting belts (62), each shifting belt having been tightened or repaired on an average only 6. times during nine years, while the cone belts averaged 32 interruptions * to manufacture in 6.7 years (61); the shifting belts having run on an average twenty-two months without tightening, while the cone belts ran only two and one-half months (56).

In ordinary shop practice, belts are tightened and repaired during working hours, and usually by the man who runs the machine which they drive; so that we are not far out of the way in considering the interruption to manufacture as about equal to the number of times each belt was repaired or tightened. When we remember, therefore, that these belts were watched

* The cleaning and greasing of the belts could readily, and probably would, be done in almost any shop outside of working hours, so that this item has been left out in 62, 61, and 56.

with unusual care, and treated in what the writer believes to be the best known manner to secure durability, he thinks it safe to say that the interruptions to manufacture in the average shop or mill would be far greater than here recorded.

68. In the writer's judgment, by far the most important matter to be determined from the above table is the reason why the shifting belts have proved themselves so much more satisfactory than the cone, and with this end in view it may be well to recapitulate the various elements which chiefly affect the life and satisfactory running of belting. They were, he believes, as follows:

(1) The material from which the belts are made and the method of their construction.

(2) The means of fastening and tightening them on the pulleys —*i.e.*, whether laced, spliced, or fastened with hooks.

(3) The care and regularity with which they are greased, and whether they are kept clean and free from machinery oil.

(4) The general nature of the service which the belts are called upon to perform.

(5) Whether the belts run vertically or horizontally.

(6) The relative length of the belts.

(7) The relative speed.

(8) The tension under which they were tightened.

(9) The average total load to which they are subjected while working.

69. Since the "shifting" and "cone" belts were made by the same makers and of the same materials, and were spliced on the pulleys and greased and treated in exactly the same way, we need not consider the first three elements as given above.

As to the fourth—"the general nature of the service of the belts"—I think that to be used on either tight and loose pulleys or cone pulleys is about as severe a duty as belts are called upon to perform. It would be difficult to say, however, which of the two duties is the harder on belting, so that I do not think that we can look here for an answer to the question.

70. Undoubtedly, horizontal belts will run under a lower tension than vertical, and in this respect the service of the cone belts was rather more severe than that of the others.

It will be remembered, however, that when the machines were moved from the old shop to the new one in which this experiment was made, the tight and loose pulleys on the countershafts

were all thrown away and far larger ones substituted for them. This was done because the shifting belts in the old shop were such a great source of trouble. They were there run under the same "total load" as the cone belts, and owing to their inaccessibility undoubtedly wasted more time than the others. It would appear, therefore, that the direction in which the belts run does not seriously affect their durability.

71. In the sixth element—"the length of the belts"—the shifting belts unquestionably have an advantage, since they are one-half longer than the cone belts (26); and it is safe to say that (other things being equal) the life of the belt should be greater as the belt increases in length, up to, say, 50 feet. The small difference in the length of the belts, however, would not at all account for the great superiority of the shifting belts, and this advantage, it would seem, must be much more than overbalanced by the fact that the shifting belts ran about twice as fast as the cone belts (29), and were always running when the main line of shafting was moving. The fact that the shifting belts ran about twice as fast as the cone belts, and yet lasted so much longer and worked in every way so much more satisfactorily, shows that the relative speed of belts (at moderate speeds at any rate) has but little effect on their durability.

72. It is, however, without doubt in elements 8 and 9 that we find the answer to our question, for while the two classes of belts were repeatedly tightened to the same initial tension, 71 lbs. per inch of width, or 239 lbs. per square inch section (see 46 and 47), the shifting belts were, owing to their greater speed and width (see 27 and 29), subjected to a far lighter total load than their competitors (see 54 and 55), and to this fact their great superiority over the cone belts is undoubtedly due.

73. The good effects of the lower "total load" to which the shifting belts were subjected can be clearly traced through several items in the table. The most important of these would seem to be item 48, showing the tension to which the belts had fallen when they required tightening.

The averages in this item * (for the cone belts especially) show the exact point in the tension of the belts at which they

* The writer did not appreciate the importance of this item when the experiment started. It was entered on the records merely as a matter of interest and without regard to its usefulness.

were just able to do the work required of them, and it will be remembered that this point was in no way arbitrarily determined, but was fixed by each belt for itself by giving indications that it was about to slip seriously. It should be noted that in the case of the shifting belts they rarely, if ever, slipped.

They were taken up, in almost every case, because they had stretched so much that they began to flop and jump about to such an extent as to be in danger of damaging themselves.

In examining items 46, 48, and 56, we note that it only took two and one-half months for the cone belts to fall from the initial tension of 71 lbs. per square inch of width to 33 lbs., their slipping point, while the shifting belts were twenty-two months in falling from 71 lbs. to a tension of 21 lbs.; and undoubtedly, if the cone belts had been subjected to as light a total load as their competitors, they would have run equally long without requiring to be tightened. Another good effect of the light "total load" on the shifting belts is seen in items 57 and 58, as they stretched $\frac{81}{100}$ of one per cent. before having to be tightened, while the cone belts, which were subject to twice as great a total load (see 54), required tightening after they had stretched $\frac{47}{100}$ of one per cent.

Summarizing the above, we may state that the total life of belting, cost of maintenance and repairs, and the interruptions to manufacture caused by belts, are dependent upon (1) the "total load" to which they are subjected, more than upon any other condition; and that, in our judgment, the other conditions chiefly affecting the durability of belting are:

(2) Whether the belts are spliced, or fastened with lacing or belt hooks.

(3) Whether they are properly greased and kept clean and free from machinery oil.

(4) The speed at which they are run.

74. Having demonstrated the fact that the life, cost of maintenance, and the interruptions to manufacture are chiefly dependent upon the average total load to which belts are subjected, it becomes of the utmost importance for us to determine the maximum average "total load" per inch of width of belt compatible with economy.

Before entering into the discussion of this matter, the writer wishes to state that he knows of no way in which this problem can be solved with absolute certainty. The conclusions finally

arrived at are dependent upon so many conditions that while they appear to be close to the truth, he holds himself ready to again modify his views on this subject, as he has several times in the past. One thing, however, is proved beyond a doubt by these experiments ; namely, that the ordinary rules place belting under entirely too severe a "total load."

75. Probably the most accurate way of arriving at the average strain to which the belting was subjected would have been a series of dynamometer tests on the machines. This, however, the writer has had no opportunity of making.

It would seem as though the effective pull of the belt on the rim of the pulley might be calculated by considering the tension of the cone belts at the time when they required tightening (see 48), in connection with their coefficient of friction, and the well-known belt formulæ.

However, in looking through the careful experiments to determine the coefficient of friction made by General Morin; Messrs. Briggs and Towne, of the Yale & Towne Manufacturing Co., Conn.; Messrs. Wilfred Lewis and J. S. Bancroft, for Wm. Sellers & Co., of Philadelphia; Prof. G. Lanza, of the Massachusetts Institute of Technology; and Mr. Edward Sawyer, of Boston, we see that the coefficient of friction varies from 15% to 135%, according to the nature and condition of pulleys and belts, the belt dressing, the amount of slip, and even the atmospheric conditions.

And, again, the notable discovery of Messrs. Lewis and Bancroft (whose experiments, on the whole, appear to be the most carefully and thoroughly conducted), that the "sum of the tensions on both sides of the belt does not remain constant," upsets all previous theoretical belting formulæ.

When we consider these facts, it would seem doubtful whether any calculation on the subject would lead to reliable results.

76. The writer, however, made out a series of tables fixing piece-work prices for almost every machine in the shop, and before arranging these tables a series of experiments were carried out to determine the pressure required to cut steel under various conditions.

Figuring backward through the driving mechanism of the machine from the cutting tool to the cone pulley, the work called for in the piece-work tables demanded an average effective pull on the rim of the pulley of 65 lbs. per inch of width of

the belt, and as the cuts called for in the tables were almost universally taken, up to the time that the belts required retightening, it is highly probable that the "cone" belts averaged an effective pull of 65 lbs. per inch of width. If this is a fact, it of course follows that the average net working load of the shifting belts is 26 lbs. per inch of width. (See 10, 18, 50, and 51.)

77. In estimating the average tension in pounds per inch of width, as given in 52, we have concluded that since it required only two and one-half months (see 56) for the cone belts to fall in tension from 71 lbs. (see 46) to 33 lbs. (see 48), therefore the fall in the tension of the shifting belts from 71 lbs. until the tension was in the neighborhood of 33 lbs. must also have been rapid. We note, however, that it required twenty-two months for the shifting belts to fall in tension from 71 lbs. to 22 lbs. It is safe to say, therefore, that during the greater part of this twenty-two months the tension of the shifting belts must have been below 33 lbs., and approaching, during this time, more and more slowly the lower limit of 21 lbs.

On drawing a curve which approximately represents the fall in tension corresponding to the months of service, the average tension of shifting belts was seen to be about 28 lbs. per inch of width (see 52), while the corresponding figure for cone belts was believed to be about 46 lbs.

Adding together the net working load (see 50) and the average tension (see 52), we arrive at the "average total load" on the belts (see 54 and 55), 54 lbs. per inch of width with shifting, and 111 lbs. with cone, belts.

78. The various authorities which the writer has studied place the safe total load of belting per square inch of section at from 290 to 500 lbs. per square inch of section. Generally speaking, these authorities base their total load upon the tensile strength of leather, taking an arbitrary percentage of this tensile strength as their safe load. So far as I know, they give no reason why they choose this particular fraction of the breaking strength as their safe load, and therefore their conclusions seem arbitrary and unsatisfactory.

79. Since the shifting belts have proved themselves so much more economical than their competitors, and since their economy is due to their lower "total load," we see, from items 54 and 55, that a total load of 111 lbs. per inch of width, or 358 lbs. per square inch section of belts, is too great for economy; 54 lbs. per

inch width, or 174 lbs. per square inch section, being far more economical and satisfactory.

Evidently then the most economical total load for belting must lie between 174 lbs. and 357 lbs. per square inch of section of belt.*

The above experiment does not, however, furnish sufficient data for accurately determining the most economical total load for belting, although it fixes two limits between which this load must lie, and I think demonstrates beyond question that the hitherto assumed "economical total loads" are entirely too high. The writer is, however, obliged, as all previous experimenters on belting have been, to arbitrarily state what he believes this economical load to be.

For several years past the writer has used the following rules with satisfaction, and he believes them to represent the most economical practice:

80. The average total load on belting should be 200 to 225 lbs. per square inch section of belt.

81. Six- and seven-ply rubber belts, and all double leather belts except oak tanned and fulled, will transmit economically a pull of 30 lbs. per inch of width to the rim of the pulley.

82. Oak tanned and fulled double leather belts will transmit economically a pull of 35 lbs. per inch of width.

83. The most economical speed for belting is 4,000 to 4,500 feet per minute.

84. The writer has recently had a most unfortunate but interesting opportunity of comparing the ordinary rules for belting with the above rules.

For three years past he has been engaged as general manager of a company in building and organizing two large sulphite pulp mills, in which about 3,500 H.P. is transmitted by belting, this power being transmitted in units varying from 1,000 H.P. to a belt down to a few horse-power per belt. The shafting runs night and day throughout the week from Sunday to Sunday, without a shut-down. Through a misunderstanding, during the absence of the writer at one of the mills, one half of the other mill, aggre-

* A total load of 174 lbs. is evidently lighter than is required for economy, since an examination of the records of the shifting belts shows that they ran on an average twenty-two months without requiring tightening (see 56), and that even at the end of this time they did not often slip on the pulleys, but were generally tightened on account of their flop.

gating about 900 H.P., was belted under the following rule, which is believed to be about an average of the ordinary belting practice:

"A double leather belt, or six- to seven-ply rubber belt, will transmit 65 lbs. pull per inch of width to the rim of the pulley."

The speed of the belting was from 5,000 feet to 6,000 feet per minute.

About one-third of the belting in the mill was leather, and two-thirds rubber.

On the return of the writer to the mill, the second half of the mill was belted on the basis of 30 lbs. pull per inch of width of double belt transmitted to the pulley, and a belt speed of 4,000 to 4,500 feet per minute.

85. It is safe to say that the belting of the first half of this mill (ordinary rule) gave one hundred times as much trouble as that of the second half. In fact, the belting proved to be the chief source of trouble and expense in running the first half of the mill, owing to the frequent interruptions to manufacture caused by it; while that of the second half ran from the start with almost no trouble.

This has proved to be a most emphatic though expensive confirmation of the results of the nine years' experiment above described.

86. Regarding the speed of belting, we see that, at moderate speeds, surely the speed of the belting has little or no effect on its durability, since the shifting belts, which lasted so much longer than the cone belts, ran about twice as fast. (See 29.)

When the speed, however, becomes sufficiently high for the centrifugal force in rounding the pulleys to play an important part in increasing the "total load" of the belt, it becomes a most important element affecting the life of the belting.

87. The writer has adopted a speed of from 4,000 to 4,500 feet per minute as the most economical belt speed for several reasons, one of which is that belting running above this speed has a great tendency to run in waves on the slack side, and flop about and oscillate from side to side of the pulleys, and so cause rapid wear, this tendency being greater and more dangerous as the width of the belt with relation to its thickness increases.

The principal reason for adopting this speed, however, is that, when the centrifugal force is taken into consideration, a total load of from 200 to 225 lbs. per square inch section gives a

maximum of efficiency when the belt runs at 4,000 to 4,500 feet per minute.

88. In this connection, I wish to call attention to the remarkable formula, developed by Mr. Wilfred Lewis, for determining the maximum economical speed of belting corresponding to given total load.

The writer regards this as one of the few valuable belting formulæ deduced exclusively from theoretical considerations, and as its value appears to have been rather overlooked, he takes the liberty of quoting from Messrs. Lewis & Bancroft's experiments, made for Wm. Sellers & Co., as follows:

" V = velocity of belt in feet per second.
S = working strength of leather in pounds per square inch.

"The velocity at which the maximum amount of power can be transmitted by any given belt is independent of its arc of contact and coefficient of friction, and depends only upon the working strength of the material and its specific gravity.

"From equation we obtain for the maximum power of leather belts the condition:

$$V = \sqrt{28\,S};$$

and for any other material whose specific gravity is y, we find

$$V = 5\sqrt{\frac{S}{y}}."$$

If we insert in the above formula the total load of 200 lbs., which the writer believes to be most economical load, we have 4,500 feet as the speed of maximum efficiency of the belt.

STRETCH.

89. A study of the stretch of belts and the frequency with which they require tightening, and the portion of their life during which they require repairing, leads us to the conclusion that there is an elastic limit in belting to a certain extent analogous, as far as it affects the life of the belt, to that existing in iron and steel. From the standpoint of stretch it would seem as though the life of belts could be divided into three periods:

(1) The period of unequal stretch, during which the rapidity of stretch and frequency of tightening gradually diminish until the

stretch becomes approximately uniform. This period lasts from about four to twelve months after the belt first goes into use; the stretch during the first week being usually very great, and gradually diminishing until it becomes comparatively uniform.

(2) The period of uniform stretch, lasting until the belt requires more or less frequent repairs.* At the end of this period some portions of the belt appear to be strained beyond their elastic limit.

(3) The period of uneven stretching. During this period portions of the belt appear to stretch much more than others, or possibly the fibres of the belt begin to yield, causing more and more frequent repairs, and finally throwing the belt out of use.

It would seem as though, for each type of belting, each belt could stretch a certain per cent. of its length before reaching its elastic limit, and that in this way the amount of stretch of a belt at any particular time might be a fairly accurate index as to its remaining life.

In examining the percentage of stretch of the shifting belts as compared with that of the cone, given in 44, it would appear that the shifting belts were not quite half worn out; and this would seem from an examination of the belts to be the case.

90. A consideration of items 33 and 34 leads us to inquire, "When is a belt worn out?" It would seem as though in some cases it would have been cheaper to put on a new belt than to continue to repair the old one.

In almost all cases a large part of the cost of repairs occurs during the last three years of its life, although with the cone belts the cost of maintenance during the first year is also heavy, owing to the excessive stretch when they first go into use. (59, 60.)

91. Referring again to the tabulated results of the leather belt experiment, next in interest to the comparison of the shifting and cone belts comes the relative merits of the various types of belts used. The oak tanned and fulled belts here show themselves beyond question to be superior in all respects, except the coefficient of friction, to their competitors.

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* Of course, the repairs here referred to are those resulting from the yielding of some portion of the leather or the splices of the belt through natural use. There are more or less repairs to every belt from the time it first goes into use, resulting from accidents.

It will be noted that a far larger per cent. of the oak tanned and fulled belts are still in use than any of the others (see 30); that their life is longer, and that none of them stretched unevenly and were thrown out of use in the early part of their life (see 40, with foot-notes); that the cost per year of service is less (see 41), the total stretch is less (see 43 and 44), and the interruptions to manufacture were less (see 61 and 62); that their competitors required tightening fully twice as often (see 56), and that, in spite of this, the average total stretch and percentage of stretch between tightenings was greater with the other belts than with the oak tanned and fulled. (See 57 and 58.)

92. It is also seen that the oak tanned and fulled belts stretch far less actually and relatively to their total stretch during the first six months of use than the others (see 59 and 60), so that they give less trouble when machinery is first started, which is the time when one can least afford to be bothered with belts.

93. We note, however, that the coefficient of friction of the other belts is somewhat greater than that of the oak tanned (see 48), since the oak tanned and fulled required tightening when their tension had fallen to 37.6 lbs. per inch of width, while the others, on an average, fell as low as 31.3 lbs. per inch of width before requiring tightening. For this reason, the journal friction, and therefore power lost, with the oak tanned and fulled was slightly greater than with the others.

94. The coefficient of friction of one of the raw-hide belts used in this experiment was determined by Messrs. Lewis and Bancroft in their experiment (the results of their experiments on this belt being given in their Table III.). The coefficient of friction of this belt, when tested immediately after its temporary removal from the pulleys, in the Midvale test, proved to be in many cases over 100%, and yet, in spite of this great advantage, this type of belting is shown in the table to be far less economical than the oak tanned and fulled.

95. The chief characteristic in which the oak tanned and fulled belts differed from the others was their stiffness. They were far less pliable than any of the others, and I am inclined to attribute a considerable part of their superiority to this property. Certainly, lateral stiffness is a most desirable quality in a belt which is to be used as either a shifting belt or on cone

pulleys. A pliable belt used under these conditions is far more apt to curl up on the edges, and to climb and stretch against the side of the cone, and finally twist up like a corkscrew. A large proportion, also, of the so-called accidents to belting, in which they jump from one cone to another, or run into neighboring gears, are due to excessive pliability.

96. Owing to their greater lateral stiffness, thick belts are much to be preferred to thin ones. So much do I believe that the property of stiffness increases the life of belts, that I make it a rule to use as thick a belt in all cases as the diameter of the pulleys will permit.

97. A manifest advantage of belts made of two or more thicknesses of leather lies in the fact that any local imperfection of the leather will produce but little effect in a double or triple belt, while in a single it is fatal.

98. Messrs. Lewis and Bancroft have, in their experiments, demonstrated the fact that "no marked difference could be detected in the power required to run a wide double belt or a narrow light one for the same tension at moderate speeds." And again, we see ropes up to two inches in diameter transmitting power with great efficiency, and with apparently but little loss of power owing to their thickness. Therefore a thick belt will be practically no less efficient than a thin one on account of its stiffness.

99. Many experimenters have shown that the pulling power of belting for a given arc of contact is almost independent of the area of the belt in contact with the pulley, and that it depends chiefly upon the sectional area of the belt, and its total tension; so that a triple belt will transmit about as much power as a single belt three times its width.

100. With wide belts, and belts running at high speed, it is especially desirable that the thickness should be increased. If thin belts are used at high speed, they almost invariably run in waves on the slack side, particularly if the load which they are transmitting changes suddenly. These waves frequently continue in the belt while it is rounding the driven pulley, so that one can sometimes even see light in places between the belt and pulley rim when standing in the proper position. This wrinkling of the belt, and the snapping that occurs as the waves straighten out, wears it very fast, and causes the splices to part, frequently in a few months. The remedy for this

trouble I have invariably found to be an increase in the thickness of the belt. When a sufficient thickness is used, the belt settles down on the same pulleys and under the same conditions to a long, steady curve on the slack side, and the wrinkling and snapping cease.

101. It would seem also as though a certain ratio of thickness to width of belt should be maintained, particularly in high-speed belts, otherwise the belt is apt to chase from side to side on the pulleys. This chasing would seem to be due chiefly to the oscillation of the belt around its longitudinal axis on the slack side, the belt being thereby tightened, first at one edge and then at the other, each side as it is tightened tending to run toward the centre of the pulley. This oscillation, and the resultant chasing, are almost sure to cease when the thickness of the belt is increased in proper proportion to its width.

102. As an illustration of this principle, the writer has in mind the case of a belt 78 inches wide and $\frac{9}{16}$ inch thick, running about 5,500 feet per minute, which could never be prevented from chasing from side to side on its pulleys for any length of time without the use of an idler pulley. This chasing was due to the oscillation about its longitudinal axis, which was caused by the small thickness of the belt in relation to its width.

A belt $\frac{7}{8}$ inch thick and 72 inches wide, used on the same pulleys, was almost entirely free from the chasing, and I am convinced that an increase to $1\frac{1}{8}$ inches in thickness would have rendered it sufficiently stiff to permanently remove the trouble. It should be noted that the thicker belt proved to be far more economical, durable, and satisfactory in every way than the thin belt.

103. If the principle is correct, of using thick belts on account of their lateral stiffness and consequent durability, it becomes of the utmost importance to determine the minimum diameter of pulley which can be used with a given thickness of belt, and still have the belt last well. The writer is quite sure that double leather belts $\frac{3}{8}$ inch thick will last well and give excellent satisfaction on pulleys as small as 12 inches in diameter, as he has had many belts in use for years under these conditions.

For some time past he has had a triple leather belt 12 inches wide, 0.56 inch thick, running about 4,500 feet per minute, with an idler pulley pressing lightly upon it, and transmitting about

100 H.P. to a pulley 12 inches diameter. This belt has up to date given excellent satisfaction, and has already lasted much longer than the two double leather belts which preceded it.

The writer feels certain, from his experience, that it is safe and advisable to use—

A double belt on a pulley 12 inches diameter or larger,
A triple belt on a pulley 20 inches diameter or larger,
A quadruple belt on a pulley 30 inches diameter or larger;

and it his opinion that it is advisable to use double, triple, and quadruple belts on pulleys respectively as small as 9 inches, 15 inches, and 24 inches diameter.

104. Regarding the question of fastening the two ends of the belt together, I think it safe to say that the life of belting will be doubled by splicing and cementing the belt, instead of lacing, wiring, or using hooks of any kind. When belts are subjected to the most severe usage, the spliced portion should be riveted, iron burrs being preferable to copper.

105. For double belting, the rule works well of making the splice for all belts up to 10 inches wide, 10 inches long; from 10 inches to 18 inches wide the splice should be the same width as the belt, 18 inches being the greatest length of splice required for double belting.

106. When idler pulleys are used on belts, the V-splice should be adopted. It is also advisable to use the V-splice on all triple and quadruple belts.

107. With rubber belts the stepped splice should be used, the spliced portion of the belt being made one or two plies thicker than the balance of the belt.

Mr. T. Robins, Jr., of New York, has recently invented a method of coating the surfaces to be spliced with uncured rubber in solution, and vulcanizing the spliced portion of the belt after it is in place on the pulleys by means of steam-heated clamps. This process, although troublesome, would appear to make the splice about as strong as any other part of the belt.

108. In establishments such as, for instance, paper and pulp mills, which should run throughout the week, night and day, without a shut-down, each important belt should be supplied with an idler pulley, which, in case a belt becomes slack in the middle of the week, can be tightened onto the belt while it is

running, and so enable it to drive until Sunday, when it can be retightened or spliced.

109. The best location for the idler pulley on high-speed belts is on the slack side of the belt, and about one-quarter way from the driving pulley. In this position it wears the belt far less than if placed close to the driven pulley, as is customary; and the tendency of the idler to guide the belt off the pulley, in case it is slightly misplaced or the belt stretches unevenly, is far less. The writer is aware that this is contrary to the accepted theories on the subject, and has only arrived at this conclusion after repeated trials.

110. The most satisfactory method of tightening belts, where the conditions permit, is to move one pulley away from the other. Countershafts, therefore, when practicable, should be mounted on frames which can be raised in tightening the vertical or diagonal belts, packing pieces of varying thickness being inserted between the frames and the supports overhead.

111. Belts drive most satisfactorily when their shafts are 20 to 25 feet from centre to centre. If they are farther apart than this, the slack side is apt to flop about too much; and if the distance between centres is much less than 20 feet, they require tightening oftener than is profitable.

With main driving belts it is often desirable to use a larger distance from centre to centre than the above, so as to secure a greater arc of contact on the small pulley.

112. The faces of pulleys should, where practicable, be made about one-quarter wider than the belts which run on them, to allow for possible uneven stretch or running of belt, and a certain amount of chasing.

113. In establishments in which the shafting is run night and day by a single motor, it is frequently desirable to stop one or more sections during the night without stopping the rest of the shafting.

In such cases it has been customary to use a friction cut-off coupling, or a friction clutch pulley, which is thrown in or out, as the case requires.

These appliances are, however, owing to the number of their parts, more or less complicated, and are certain, sooner or later, to get out of adjustment and cause more or less trouble and repairs.

The writer has found a far more elementary and satisfactory

contrivance for this purpose to be a pair of tight and loose pulleys, ~~onto~~ either of which the driving belt can be readily shifted while it is running at full speed, and he has had, during the past two years, lines of shafting transmitting as high as 300 H.P. successfully thrown off and ~~onto~~ the main line each night and morning.

114. The two elements in this apparatus which are likely to cause trouble are, first, the loose pulley, and, second, the belt shifter. The loose pulley which I have found most successful for this purpose * has a loose bronze bushing in its hub, which is perforated with holes, and is automatically deluged with oil each time that the pulley is stopped.

115. The writer has used successfully, for shifting large belts at high speeds, two styles of belt shifters, both of which are old in principle, but which—so far as he knows—have heretofore been but little used. These belt shifters are, as usual, applied to the belt on the slack side, and as close as practicable to the driven pulley.

116. The first consists of a pair of rollers mounted on a movable frame, and having their axes inclined toward one another, forming an angle of about 20°. This frame is then placed so that the plane of the axes of the rollers is at right angles to the centre line of the belt, and the belt passes between the rollers without touching either. Belts up to from 20 inches to 24 inches in width are readily shifted by pressing one or the other roller against the edge of the belt, the action being similar to that of an ordinary forked belt shifter, except that the edge of the belt curves downward, owing to the inclination of the rollers, until the body of the belt touches the roller, and in this way the edge is prevented from burning and fraying out.

117. The second style will readily shift any width of belt whatever, and for wide belts, when space permits, is to be preferred to any other style of shifter. Two rollers, the length of each of which is rather more than twice the width of the belt, are placed with their axes parallel to the plane of the belt, each forming an angle of about 75° with the centre line of the belt. They are inclined, however, in different directions to the centre line of belt. When one of these rollers is pressed even with a moderate force against the surface of the belt, it will move

* This pulley is made by John McCaffrey, of Lawrence, Mass.

slowly and uniformly to the right until the roller is removed. If the other roller is used, it moves in the same manner, but in the opposite direction.

The action of this belt shifter, in starting a line of shafting into motion, is much to be preferred to that of any friction clutch, it being exceedingly uniform, smooth, and free from jar.

118. It is undesirable to run a loose pulley for any length of time, however, under heavy belt pressure. Each head shaft, on which the tight and loose pulleys are mounted, should be coupled to the line which it drives with a flange coupling, having a plate about $\frac{3}{4}$ inch thick between the two flanges. When it is desired to stop the line of shafting for several hours in succession, the belt should be temporarily shifted onto the loose pulley, the flange coupling should be opened and the distance plate removed, and the belt shifted back onto the tight pulley where it can run without danger from cut or hot bearings.

A starting and stopping device of this sort is as durable and gives as little trouble as any part of the shafting.

119. As shifting belts are ordinarily used, they are injured as much by running partly on the tight and partly on the loose pulley as from any other cause. It is, therefore, of great importance that each ordinary forked belt shifter should be provided with some appliance for insuring the belt being either entirely on the tight or the loose pulley.

We have found a simple and effective plan to be that of cutting two V-shaped notches in the edge of the slide to which the forks are attached, the same width each at the top as the face of the pulleys, and pressing a male V down into these notches with a light spring, the sides of the notches being made so steep that the male V could rest nowhere but at the bottom of one or the other of its mates.

120. Belts should be cleaned and greased every five or six months, just enough grease being put on to keep the surface of the belt moist and prevent it from cracking. It was found in the above experiment (see 13) that every three months was oftener than belts required greasing.

121. Belts will last well if repeatedly tightened under a strain of 71 lbs. per inch of width of double belt, equivalent to 239 lbs. per square inch of section. (See 46 and 47.) It is evident, however, that they will not maintain this tension for

any length of time while working, since, as shown in the table, they fell to one-half this tension in about two and one-half months. (See 48, 49, and 56.)

122. It is very important for the life of belts, as well as to avoid excessive friction in the bearings, that they should not be overstrained (see 68 to 73); therefore, where the conditions permit, belt clamps should be used which are provided with spring balances for weighing the exact tension to which the belt is tightened, and a table should be carefully figured, indicating the strain to which each belt in the establishment should be tightened.

In many cases, however, it is impracticable to use spring-balance belt clamps in tightening belts, and in such instances it has generally been customary to guess at the amount of belt to cut out; and the results of this guessing often prove disastrous both to belts and bearings.

On examining items 57 and 58 in the table, we note that the cone and shifting belts stretched respectively $\frac{47}{100}$ and $\frac{81}{100}$ of one per cent. of their lengths between the times of their tightening, and from this we deduce the following rough rule as an aid in determining the proper amount to shorten belts:

It is safe to shorten a double belt $\frac{1}{2}$ inch for every ten feet of length, if it requires tightening, when working according to the ordinary rules under a total load of 111 lbs. per inch of width, and giving an effective pull of 65 lbs. per inch of width.

If it works under a total load of 54 lbs. and effective pull of 26 lbs., it is safe to shorten the belt one inch for every ten feet of length.

123. Serious repairs to belting, as well as to all other machinery in a mill, should be prevented as far as possible by systematic and careful inspection at regular intervals, and the writer has found a tickler, having a portfolio for every day in the year, from which reminders to inspect and examine are issued daily, an invaluable aid in caring for the machinery of an establishment. With this method, a belt should rarely slip or give out while in use, and most repairs can be made out of working hours.

124. Much time is saved by having all of the repairs and adjustments to belting made by one or two men. A day laborer can soon be taught to repair belting after working hours, and do it much more thoroughly and systematically than if it is attended

to by the high-priced men who run the machines during working hours.

125. In figuring the probable running expenses of an establishment, it is frequently desirable to know about what the yearly belt bill will average.

Items 32 to 41 inclusive cast some light upon the subject, and we find this information summarized in item 42. Here we see that if belts are tightened according to the ordinary rules, so as to drive 65 lbs. per inch of width of double belt, the cost of all labor and materials used in their maintenance and repairs, added to the cost of renewals as they give out, through a term of years, will amount per year to 37% of the original cost of the belts. These belts were treated with exceptional care, so that in most establishments the cost would be greater than this.

If, however, the belts are tightened so as to transmit 26 lbs. per inch of width, the annual cost of maintenance and renewals only amounts to 14% of the first cost, and probably less.

DISCUSSION.

Mr. Henry R. Towne.—This paper is of such unusual character and contains so much of value that I am prompted to briefly discuss some of its important features, my interest in its subject dating from 1868, when, at the instance of my friend, the late Robert Briggs, I undertook a series of experiments to determine the coefficient of friction of belting.

The present paper is modestly entitled "Notes on Belting," but could more fittingly be described as a treatise on the practical use of belts. Its thirty-four pages contain more new and useful information relating to the efficiency and economy of belts than is to be found in any other paper which has come to my knowledge. The paper is further remarkable as covering the record of an unusually large series of experiments, inaugurated on an intelligent and exceptionally comprehensive plan, and subsequently consistently carried out during the extraordinary period of nine consecutive years, under conditions not of the laboratory but of actual practice.

An excellent feature of the paper is the mode of presenting the results of the experiments it records, which is so condensed and pithy as to make it difficult to discuss its contents briefly.

The text of the paper is so full of pertinent and suggestive facts, in such condensed form, as to make it easier to quote from it than to criticise or review it. Perhaps the most salient fact, and the most important conclusion of Mr. Taylor's argument, relates to the value of *increased thickness* of belts, and to the larger and more general use of double belts. Whereas formerly it was supposed that pliability was desirable, and that therefore wide thin belts were better than narrow thick ones, the paper asserts the contrary, explains the reasoning on which this assertion is based, and adduces the evidence of experience to prove it.

Having established this point, the paper then presents rules, and the experience on which they are based, governing the conditions under which thick belting can most efficiently and economically be used. These rules pertain less to the theory than to the practice of belting, and cover the questions of speed, diameter of pulleys, modes of tightening, distance of pulleys apart, kind and frequency of dressing, methods of lacing, and the efficiency of different kinds of belting. Previous investigations have dealt rather with the theory of belting than with the question of economy in its application and use. One of the most valuable features of Mr. Taylor's work consists in determining the conditions of application and use of belting which conduce to the lowest ultimate cost. In other words, his investigation carries the subject through the field of mechanics into that of economics, and reduces the equation finally to a commercial form. However much the theoretical questions involved may interest the student and engineer, the commercial facts are those which chiefly interest and concern the mill manager and owner. In this, as in similar cases, the work of investigation is not completed until it has been carried to a point which includes both mechanical and commercial factors.

Mr. Taylor's conclusions are based upon the record of observed facts gathered during a period of nine years, and relating to the belting used under various conditions of actual service in the works of the Midvale Steel Company. The test was projected when the system of belting and shafting was designed. The experiments have thus been based on certain predetermined conditions, which have been consistently maintained from the beginning to the close of the period of observation. These conditions are those of ordinary practice, and, providing they cover

a sufficient range of cases to constitute a fair average, are better for the present purpose than the conditions existing in the case of laboratory experiments, which form the basis of discussion of most other writers on this subject.

The work of previous investigators has related chiefly to determining the coefficient of friction. Noting the wide discrepancy in recorded observations on this point, Mr. Taylor has wisely refrained from further investigation of it, and has directed his observations and reasoning to the other elements involved, and for the purpose of arriving at rules for obtaining the best practical and commercial results. The records show that the coefficient of friction is itself variable, and depending upon the kind of belt, the condition of its surface, and other variable factors. Hence it follows that, in most cases, it is practically unnecessary to consider the coefficient of friction at all, the other more important factors which determine the proper conditions of use, especially those conducive to the best economy, and which determine the transmitting efficiency, being such always as to preclude any slipping of the belt.

Mr. Taylor attaches great importance to the question of maximum tension, and urges the adoption of methods which will restrict this within certain definite limits, which he has endeavored to determine. The importance of adopting means for determining and limiting the tension of belting is indicated in the statement in the paper, based on observed facts, that the maintenance and repairs of belts tightened according to the ordinary rule amounted in one year to 37 per cent. of the cost of the belts, while this cost was reduced to only 14 per cent. when the belts were tightened according to Mr. Taylor's rules.

Mr. Taylor properly gives much consideration to the question of economy in time of men and machines by using belting in such way as to secure the maximum freedom from interruptions to manufacture due to this source. In this, as in other details, his rules aim to secure results which are not only the best mechanically, but also commercially. The evidence submitted sustains the assertion that to obtain the highest economy in belting it is necessary to limit both the *initial tension* and the *total load* to a point much below that fixed by former rules, and which Mr. Taylor has sought to deduce from his observations. To accomplish this reduction he resorts to the use of double belting, adduces facts tending to show that this can be used on

pulleys as small as twelve inches in diameter, and shows that, for most uses, double belting is not only as available as single, but in many respects better, more desirable, and more economical.

He sums up his conclusions by the statement that the most important factor in determining the life of belting, and its cost for maintenance and repairs, is the *total load* to which it is subjected, meaning thereby the pull per square inch of cross section. Other things being equal, therefore, the life and economy of the belt will, up to a certain limit, vary directly with its thickness. The conclusions on this point are that the point of best economy of total load lies between 174 lbs. and 357 lbs. per square inch of cross section. Mr. Taylor states that he has adopted a limit from 200 to 225 lbs. If his other conclusions and rules are adopted, further observation and experiment may well be addressed to a closer determination of this factor, in order to either verify the correctness of Mr. Taylor's conclusions, or else to indicate a new and better value for adoption in practice.

In effect, Mr. Taylor has thus accomplished a change of base which implies that future work should follow a new line of investigation. Heretofore experimental work has been directed chiefly to determining the coefficient of friction. Hereafter it would seem that such work can be best addressed to determining more accurately the *total load per square inch of cross section* which will insure the greatest economy. This change of base is suggestive of the similar change which has occurred during recent years in investigations relating to the strength of metals. Whereas earlier investigations sought to determine their ultimate or breaking *strength*, it is now found to be much more useful to determine accurately their elastic limit. Since present practice is based upon the latter condition, and the question of ultimate strength thus becomes of minor importance, Mr. Taylor's investigations indicate that, in like manner, the limiting factor in the use of belting is the total load per square inch of cross section, and that if this condition is properly adjusted the question of friction will take care of itself, and becomes a minor issue.

In conclusion it is well to repeat again that the paper is so full of pertinent and interesting facts as to make it impossible to review it fairly without practically transcribing the paper

itself. The latter, therefore, should be read carefully and in full by every one interested in the subject and desirous of availing of the latest knowledge as a guide to practice.

Mr. William Kent.—The most notable feature in Mr. Taylor's paper is the great difference between his rules for proper proportioning of belts and those given by earlier writers. A very commonly used rule is, one horse-power may be transmitted by a single belt one inch wide running x feet per minute, substituting for x various values, according to the ideas of different engineers, ranging usually from 550 to 1,100.

The practical mechanic of the old school is apt to swear (as I have heard them do on the witness stand) to the figure 600 as being thoroughly reliable, while the modern engineer is more apt to use the figure 1,000. Mr. Taylor, however, instead of using a figure from 550 to 1,100 for a single belt, uses 950 to 1,100 for double belts. If we assume that a double belt is twice as strong, or will carry twice as much power as a single belt, then he uses a figure at least twice as large as that used in modern practice, and would make the cost of belting for a given shop twice as large as if the belting were proportioned according to the most liberal of the customary rules.

This great difference is to some extent explained by the fact that the problem which Mr. Taylor undertakes to solve is quite a different one from that which is solved by the ordinary rules with their variations. The problem of the latter generally is, "How wide a belt must be used, or how narrow a belt may be used, to transmit a given horse-power?" Mr. Taylor's problem is, "How wide a belt must be used so that a given horse-power may be transmitted with the minimum cost for belt repairs, the longest life to the belt, and the smallest loss and inconvenience from stopping the machine while the belt is being tightened or repaired?" The two problems are just as different as those we have sometimes to solve in reference to boilers and engines. If we have a saw-mill where fuel costs nothing, our boiler and engine will cost us a certain figure for a given horse-power; but if we have a cotton-mill where coal is expensive, and the boiler and engine are designed to economize coal, they will cost twice as much for the same horse-power as those at the saw-mill.

The difference between the old practical mechanic's rule of a one-inch-wide single belt, 600 feet per minute, transmits one horse-power, and the rule commonly used by engineers, in which

1,000 is substituted for 600, is due to the belief of the engineers not that a horse-power could not be transmitted by the belt proportioned by the older rule, but that such a proportion involved undue strain from overtightening to prevent slipping, which strain entailed too much journal friction, necessitated frequent tightening, and decreased the length of the life of the belt.

Mr. Taylor's rule substituting 1,100 feet per minute and doubling the belt is a further step, and a long one, in the same direction. Whether it will be taken in any case by engineers will depend upon how they appreciate the extent of the losses due to slippage of belts slackened by use under overstrain, and the loss of time in tightening and repairing belts, to such a degree as to induce them to allow the first cost of the belts to be doubled in order to avoid these losses.

I regard Mr. Taylor's paper as a most important addition to the papers on the subject of belting already printed in our Society's transactions. It records the only systematic series of experiments on belts in actual use extending over a period of years of which I have any knowledge. His conclusions, while in some cases they may be modified after longer experience, should be accepted, unless they can be overthrown by results of other experiments conducted with equal care for an equal period of time.

The question may be raised, why, if Mr. Taylor's rule (using 1,100 for the factor for a double belt) is correct, has it been found possible to do so well in practice with a belt proportioned according to the older rules, with a factor of only 550 for a single belt, which older rule would give a belt only one-fourth the sectional area of one proportioned by Mr. Taylor's rule? The answer is that in many cases belts which were designed under the old rule are really running under Mr. Taylor's rule. That is, a belt proportioned for and estimated to be driving four horse-power is really driving only one horse-power. A belt is usually, in the millwright's practice, made wide enough to drive the machine at the maximum power expected or guessed to be required at any time. If the belt were actually required to transmit the horse-power continuously, it might require tightening once a month and be worn out in two years; but having to transmit the maximum power only occasionally, for brief intervals, and averaging a power only one-fourth of the maximum, it may require tightening only once a year and last

twenty years, and be used as an eminent example of the merits of the old rule.

It is to be hoped that since Mr. Taylor has shown how to make a nine years' test of belting, that others of our members will carry on a similar line of experiments, and that ere many years we may be able to collate the results into a series of rules that may find general acceptance and put to an end the confusion in which the subject of belt proportioning now rests.

Mr. J. H. Cooper.—It is true, as stated by the author of these notes, that nothing seems more simple than belt problems. The mass of data here presented, and the considerable collection in *Use of Belting*, show the belt subject to be a very complex one. It may be truly said that continued and deeper study of any subject brings new properties to light, finds different relations and bearings of them, and frequently shows the necessity of revising the existing formulæ and rules of application. In this particular the "best practice" is made better by the last experimenter, if he have gone beyond the confines of existing knowledge and experience.

I have not found all belting rules based upon their maximum efficiency; very plainly there are judicious checks upon this. Within the limits of usual conjecture, it is fair to assume that with the empiric, generalizing from the narrow limits of his practice; the dealer, interested chiefly in selling the widest belts; the buyer, who looks mainly to the economies of cost, transmission, and maintenance; and the theoretic, applying his formulæ to principles of action, there would be very wide divergences of result and rule. But the exhibit in *Use of Belting* shows a rather unexpected nearness of agreement among the means of the extended list given which largely prevail in number. There is abundance of statement about them to prove that nearly all the rules have a liberal factor of safety, and do not punish the material beyond acceptable limits.

But just here I wish to emphasize the importance of Mr. Taylor's experiments, which show the effect of constant hard work on belts, and the consequent necessity of reducing the load upon them. Therefore, rules which may include occasional excessive strains must be modified to suit cases where such strains are constant; so it is again proven that the material will give its own rule if it be put to the test. This Mr. Taylor has effectually done.

The low working tension upon belts found most effective by

Mr. Taylor proves a substantial origin of those rules based upon a working tension of from 19 to 30 lbs. per inch of width, as recorded in *Use of Belting*. These rules may have been derived from lengthened actual trials, proceeding upon the ancient slow and sure plan rather than upon the modern high-pressure principle of action. Mr. Taylor's note on the effect of centrifugal force upon belts during high speeds, by which the tension must be increased to keep the belt in contact with the pulley, which in turn necessarily shortens the life of the belt, is worthy of attention. Also his query as to when a belt is worn out; how often a penny-wise-and-pound-foolish economy keeps patched-up belts, always breaking, to the loss of output of the machine as well as of the temper of the operator.

Mr. Taylor's method of tightening belts by separating their pulleys is a very good one, but cannot always be carried out in the design and installation of machines. In many cases, however, as is done with dynamos, such devices might be incorporated in the plan of the machine.

There is a general neglect in the workshop of *best* fastenings and *best* care-taking of belts, notwithstanding the abundance of published and prevailing advice about these matters. The means of "securing the least possible interruption to manufacture" by belt failures involves a position well taken by Mr. Taylor, wherein he has developed one of the less known features of the use of belting.

In reference to the weight and speed of belts, the general belief is in favor of those which are light and swift running; and to secure the best economy of them their pulleys must be properly proportioned in diameter and face to suit the needs of the belt. There must be uniformity of belt section and weight, and well-balanced parts everywhere while running.

Economy in the construction of the machines themselves, apart from the belts which drive them, may favor the machine more than the belt. The maximum output of the same may therefore throw unusual strain upon the *usual* belt with which it is supplied.

Mr. Taylor's experiments may have been upon just such machines, in which case a general rule cannot be formulated from these and like special results. There is no question, however, in the belt element, about the greater expense being that of loss of time, and the consequent loss of production of the

related machines. A belt best proportioned to the work it has to do, and if well cared for, will not be likely to fail early in its life. Mr. Taylor's experience, as related in these "Notes," proves the lack of belt power for driving machine tools, especially under severest conditions, and this may be stated as a matter of frequent observation.

Experiments with belts as usually made are of short duration, intended chiefly for ascertaining some desired phenomena of belt action, and it is also the fact that a very small number of recorded trials have been extended through long periods of time. *Use of Belting*, however, does record the experimental researches of M. Leloutre, a French engineer, who conducted an extensive series of tests of belts in actual use in the mill through a period of a dozen years or more, in connection with important works which he had in charge. These researches were upon the stretching, the elasticity, and the resistance to rupture of belts of leather, gum, and cloth; upon the slipping of belts and ropes, and upon the transmitting power of belts. This engineer also says that "the care of stretching belts should never be left to workmen;" that dynamometrical forceps should be used, by which a known tension can be put upon belts when they are laced or joined.

Much was proven by this engineer, but there yet remained to be established the commercial value of belting considered simply as an organism for transmitting force from one part of a system of machines to another. Mr. Taylor's experiments largely supply this need, covering this ground in considerable thoroughness, giving us also quite an array of carefully conducted and profusely exemplified facts and data.

The true economics of belts, as of anything else, can only be derived from the *long use*, which mostly brings in elements fraught with certain results not developed by trials conducted through short periods of time.

Mr. Taylor's belts were of best known materials and make; they were well cared for, were properly stretched and loaded. From such completeness of provision and management we should expect conclusive results, since we are certain that "nothing imperfect is the measure of anything." Mr. Taylor's experiments exhibit belts as they really are, not as one may think they ought to be; they tell of belts, much as belts would say if they told us, on the principle that "the best knowing comes from the best doing," the practical quality of a thing being that

which we must have if work is to be done; and the data of actual performance points out the shortest road to successful use.

Mr. W. S. Rogers.—I do not think the conclusions and deductions in this paper will bear adoption in all shops, at all times, and under all conditions, without serious consideration of all surroundings and financial possibilities.

As the author has confined his paper to machine-shop practice, it would be unfair to go outside of this field, so that I confine my remarks closely to the conclusions from his nine years' experience. His statement in article 20, that the records of many of the individual belts are erratic, so that it is only by a series of averages that we can arrive at any reliable conclusion, is such a serious contradiction of the statement in article 15, as to an accurate account of every item of expenditure, as to throw discredit upon the safety of applying his rules in any given case. The special problem before the engineer may turn out to be a specimen of an erratic case.

I also want to dispute the author's statement that in starting new shops serious hitches are always to be found. I do not think this is fair to our machine builders.

I have never found any difficulties whatever, and I have never done anything but fool around machinery all my life. I have started several new shops, and have had the pleasure of seeing the inside of perhaps ten thousand others while rambling over our country, and I never hear of any hitches or troubles, provided the men in charge of it recognize the fact that it is a new infant, and care must be used until "the corners are worn off," and that it cannot be put into hard service at once. No railroad man would think of putting a locomotive just built into fast passenger service. If he did he would have a "hitch." In regard to the time lost in shops by belts giving trouble, I will admit that the saving of time is the one great object of importance in all shops, but the great percentage of time lost and wasted in manufacture is not traceable to troublesome belts and machinery and workmen, half so much as it is to foremen, superintendents, and managers devoting too much of their time to other matters, instead of seeing that dividends are earned, and the men under them led on to greater achievements and more profitable results regardless of personal hobbies.

Referring to article 64, I have found in my experience, taking the running conditions of the two kinds of belts in daily service,

without comparisons as to length as mentioned in article 65, that horizontal running shifting belts in machine-shop work usually last about five times longer than the vertical cone belts, while in wood-working plants, where the service is harder and more furious, they last only three times longer than the vertical belts. The chief reason why the shifting belts last so much longer than those used on cones is that cone belts have about three times more work to do and receive about ten times more abuse while doing it than their brethren who drive them. I know of nothing more destructive to belting in service than cone steps, and the methods used for shifting belts on them. There is one point in regard to the life of belts which we seem never to take into consideration when discussing their extended or abbreviated lives, and that is the temperatures through which they have to pass in performing their functions. I have found that a belt running through varied changes of temperatures will cause considerable more trouble and obstruction to manufacture and have a much shorter life than one running under more favorable conditions.

In article 71 it is said that "the small difference in the lengths of the belts, however, would not at all account for the great superiority of the shifting belts, and this advantage, it would seem, must be much more than overbalanced by the fact that the shifting belts ran about twice as fast as the cone-belts," etc. Consulting the table, I find this "*small difference*" to be slightly over 60 per cent. in favor of the shifting belts; and as it is stated in the last part of this same article that "the relative speeds of belts have but little effect on their durability," I cannot see just where the "overbalanced" argument comes in. Practically, I cannot see how the fact stated in the last paragraph of this article proves anything whatever, except that belts running under different circumstances and conditions display greater or less durability, and cause more or less trouble, as the cases may vary.

On page 223, at the top, in speaking of the exact point of tension at which belts were just able to do their work, the author says this point was fixed by each belt for itself, by giving indications that it was about to slip seriously. Now I would like to ask what that sign or indication is? The only evidence I have ever been able to gain from a belt as to its intentions to slip was after the slip took place. In this same paragraph we read that "the shifting belts rarely if ever slipped," and bearing this

in mind, I read on in the next paragraph in regard to the shifting belts, "that they were taken up because they had stretched so much that they began to jump and flop about so as to endanger damaging themselves." Now I have always found that a belt was slipping considerably when its actions were as here described, and the tension was about zero; and not only is it damaging itself, but it is also injuring the smaller belt it is to driving, a certain extent.

On page 226 the author states, in paragraphs 80, 81, 82, 83, and 84, the rules he has adopted and used for several years, and when I come to No. 83 I make a few figures to see how I would have come out in equipping my plant this autumn, provided I had followed these rules.

My three line shafts have a rotative speed of 200 revolutions per minute, which is a little above the average in machine-shop practice; to have obtained the economical belt speed called for in paragraph 83 I would have been compelled to use pulleys seven feet in diameter, and hangers having forty-five inches drop at least. As the ceiling is 16 feet from the floor (which is a foot or two more than usual practice in the great majority of manufacturing shops), the lower side of the belt would have been only 9 feet 9 inches from the floor. If I had belted to a heavy boring-mill, whose countershaft pulleys are only 28 inches diameter, and are figured to run at 125 revolutions per minute by the builders, I find that I would have required tight and loose pulleys over eleven feet in diameter, and the tool builders would have had to also furnish me with hangers for the counter having 6 feet drop. This would have brought the lower side of the counter-belt to within 5 feet of the floor; and should I have carried this plan out all through the plant, I should have had to rent space elsewhere to work in. To have run a triple-cylinder moulding and flooring machine, and carry the belt speed of 4,000 to 4,500 feet to the countershaft, whose builders call for a pulley speed of 1,000 revolutions per minute, I would have required 16-inch pulleys on the counter instead of the 10-inch ones furnished. My shafting and hangers would also have had to be much stronger to carry this excess weight, and the first cost would have given the stockholders paralysis.

Referring to the third paragraph in article 113, I must beg leave to differ most emphatically. Friction clutches, given the same care and attention that is, or rather should be, expended on loose

pulleys and belts, will give more satisfactory results on the belting and throughout the shop generally. I would much rather have the wear and burn from slow starting come on the wooden shoes of the clutches than on the belting. The wooden shoes are cheap, and easily replaced when the life is worn out of them from frictional slipping, while I know of no way of putting it back into the belt. Friction clutches are superior things in case of accidents, as has been proven thousands of times for the benefit of employer and employee, and machinery as well.

On page 234, article 112, the conclusions drawn and the reasons therefor again have set my arithmetic in motion; but before I get to figures I would like to say that a belt so poorly made or in such condition as to require 25 per cent. extra width of pulley to chase itself on would not be in service long in a plant under my control, as I take just as much pride in the appearance of things in general as I do in their utility.

The market prices in competition to-day for a large driver and tight and loose pulleys, calculated according to this conclusion, would be as follows, using an 18-inch belt:

Driver, 10 ft. diam., 10 in. bore, 50 in. face....	$400,	weight	13,000 lbs.
Tight pulley, 7 ft. diam., 25 in. face, 5 in. bore	97,	"	2,100 lbs.
Loose pulley, 7 ft. diam., 25 in. face, 5 in. bore	115,	"	2,200 lbs.
Total cost and weight	$612,		17,300 lbs.

The cost, according to general practice for the same drive, would be as follows:

Driver, 10 ft. diam., 10 in. bore, 40 in. face...	$365,	weight	10,000 lbs.
Tight pulley, 7 ft. diam., 20 in. face.....	90,	"	1,900 lbs.
Loose pulley, 7 ft. diam., 20 in. face.....	108,	"	2,000 lbs.
Total cost and weight.....	$563,		13,900 lbs.
Excess cost and weight per conclusion 4.....	$49,		3,400 lbs.

In both of these estimates the drivers have double arms, and the loose pulleys have metallic bushings.

Estimating for the same sized belt drive, and using one of the best friction-clutch pulleys on the market, we have the following figures:

Driving pulley, 10 ft. diam., 10 in. bore, 20 in. face.	$150,	wt.	6,000 lbs.
Friction-clutch pulley, 7 ft. diam., 20 in. face.....	285,	"	4,000 lbs.
Total cost and weight.....	$435,		10,000 lbs.
Saving in first cost and weight in favor of friction pulleys over conclusion 4's method.....	$177,		7,300 lbs.

The $177 would keep the clutch in repair for more than ten years, and the economy in eliminating nearly four tons of dead weight at this one point alone would be shown up daily in the furnace-room. By the way, of what real use is the advice given here without the data of coal consumption and water evaporation from the engine and boiler-rooms?

On page 231, in article 98, the comparison of rope driving with heavy belt drive, as showing the advantages favorable to thick belts, is hardly fair, I think, as the action of a 2-inch rope in service is very different from that of a flat belt.

Taking a run through the conclusions the author has drawn, I find the first one averages about the same as those in the text-books. I differ with No. 3 by thinking that 30 to 35 feet between centres is best practice. No. 4 I have already found fault with. I know of cases contrary to the conclusion in No. 7, the homeliest illustration being driving the foundry rattlers. My experience has not been so exact as to say that triple belts must always be used down to 20-inch pulleys. In truth, unless the case was exceptional, I would not use them under 42 inches. Referring to conclusion 10, I would say that I have tried this method of belt tightening, and I have also tried moving the machine away from the counter, and of the two miseries the latter possesses a little more merit, to my way of thinking. When machine builders make their countershaft hangers with adjustable bearings for this purpose, it will be well; but loosening up a counter, removing the wedges, and realigning and allowing the tension of the horizontal belt to change, in order to tighten the cone belt, is not a mechanical method. No. 14 is the old Scotch millwright's rule in another form: "An inch to every ten feet in new belt, boy, and half an inch for every ten foot afterward." Conclusion No. 34 I have previously disputed; there is nothing more convenient, neater, and easier of manipulation about a machine counter than a properly made and simple friction clutch, decently cared for.

In conclusion 2, page 205, we are told that 4,000 to 4,500 feet per minute is greatest economy; and on page 212, paragraph 3, the writer's judgment is that we should lower the speed and use heavier belts to reduce belt bills, although it is stated elsewhere that high speed shows no effect on the durability of belts; and on page 226, article 83, he states his practice for several years. Now, I would ask, what is high speed and slow speed? I don't

think that, after leaving the prime movers, there is any machine-shop that I know of whose main-line shafting is running at a velocity sufficient to give the belt speeds called for in this paper.

Mr. Robert Cartwright.—It strikes me that the author is unwise in attempting to reduce his observations from machine-shop practice into rules which he seems to suggest for universal application. We are not all running machine shops, and for such locations as damp wheel-pits, and places where the belt to-day is three inches longer than it will be to-morrow, he would appreciate the reluctance with which any rule can be accepted. I have belts under such circumstances, 24 inches wide, driving shafts 90 feet between centres, and for a case like this nothing but the belt tightener will answer at all. Even this has to be set up to-day and loosened to-morrow by reason of changes in the hydrometric condition of the air.

In paper mills, no theory for a relation between strain and durability of a belt can go for anything, by reason of the rapid deterioration and destruction of belts in the atmosphere of such mills. The lime constantly present in the air proves very destructive, and so also does carbonic acid in the air of a blacksmith shop. In both of these cases there can be no comparison with the conditions prevailing in a well-ventilated machine shop, where also there are no extremes of temperature.

The position, also, of a belt has much to do with the conditions for its satisfactory running. I have belts driving dynamos which I have had to place vertically, and in these cases a large element to be taken into consideration is the gravity weight of the belt itself, and often the speed of the belt is fixed by other conditions than a choice according to rule.

I have belts running at 4,500 feet to 7,500 feet per minute in dynamo service, and I have 5-inch shafting at 350 revolutions, and 4-inch shafting at 440 revolutions, per minute. It does not therefore commend itself to my judgment and experience to formulate rules for economical speed when an engineer is so often called to run belts in every direction, in all kinds of temperature and for all sorts of service. My belief is that every case is to be governed by its surroundings, and the judgment and experience of the engineer must be mainly relied upon rather than formulated rules.

Mr. Wilfred Lewis.—However opinions may differ as to the

wisdom of the conclusions arrived at in this paper, the value of the facts presented can hardly be overestimated. They represent an amount of labor and care over an extended period of time which is rarely bestowed upon any subject, and the conclusions reached are therefore worthy of the most serious consideration. They should not be judged hastily, nor from a different point of view than the one distinctly stated in the paper.

The author states that "in using belting so as to obtain the greatest economy and the most satisfactory results" certain rules should be observed, and it is no argument to contend that in applying these rules other sources of expense will be increased.

It is right and proper that other questions should be considered before giving way to the dictates of economy in belting, but there is no need of confounding the belting question with any other, as some of us may be inclined to do from the first glance at these conclusions.

For example, the second conclusion is a natural deduction from the first, and in fixing the belt speed for maximum economy at 4,000 to 4,500 feet a minute, no claim is made that all belts should run at that speed, regardless of other consequences. But so far as economy in the use of the belting itself is concerned, the conclusion means that all belts should run at that speed to transmit the maximum amount of power. This conclusion is certainly borne out by the facts given, and the cost of pulleys and shafting has nothing to do with it. That is another matter that must receive attention before the belt speed is finally determined, and it would be absurd to regard pulleys and shafting as the subject under discussion.

Conclusion No. 4, that the faces of pulleys should be about 25% wider than their belts, likewise refers to economy in belting and not to economy in pulleys, or the combined economy of pulleys and belting. Whether or not such an unusual increase in the width of a pulley can be afforded must frequently be determined by other considerations than that of belting; but that such an increase in width will be advantageous to the life of belting can hardly be doubted.

Conclusion 5 is the result of experience which is rarely encountered in machine-shop practice.

Conclusion 7, that belts made thick and narrow are better

than wide and thin, confirms the writer's theory that this should be the case; for if a belt ¼ inch thick runs well on a pulley 1 foot in diameter, a belt 1 inch thick should run just as well on a pulley 4 feet in diameter; and by following the advice given in conclusions 8 and 9, the space occupied by belting may be reduced below ordinary practice while still retaining the extra width of pulley called for in conclusion 4.

An objection might be made on theoretical ground to the loss of power in bending and unbending a thick belt over a thin one; but this is a small matter, which is probably more than offset by the lesser air resistance of a narrow and thick belt over a wide and thin one.

Conclusions 10 and 11 may be admitted without question, and the twelfth conclusion need not be judged by one who has not given it a fair trial.

As few persons will care to go to the trouble of weighing their belt tension, as suggested in conclusion 13, the next conclusion as to the amount of shortening is probably of more interest and value to practical mechanics.

Conclusions 15 and 16 will not be doubted, and conclusion 17 is of interest as indicating a limit to sustained tension.

Conclusion 18, stating the fall in tension to be expected under the ordinary rules for belting, and the following conclusions 19, 20, and 21, contain the pith of the argument for lower belt tensions than are generally used, and indicate the laborious character of the work leading to them.

Hitherto the load to be carried by belting has been determined from its tensile strength by some arbitrary factor of safety, while the results here given are based upon its elastic limit, if such a term can be applied to belting, and the cost of maintenance and repairs under the different loads is a matter of the greatest interest and importance.

Conclusion 22, that the total life of belting depends upon the total load carried rather than the work done, is somewhat surprising, and emphasizes the importance of light tension and high speed to attain the greatest duty.

From conclusion 23 it appears that the bending and unbending of a belt by rapid motion over pulleys has but little effect upon its life, and it must be inferred that whatever loss is found in speeds higher than 2,500 to 3,000 feet is due to some other cause than simple bending.

The whip-like action which takes place at very high speeds has no doubt a great deal to do with the wear and tear, but by thickening the belts in proportion to their width, as suggested, there seems to be good reason to believe the life of belting will be increased.

The data given in conclusions 24, 25, and 26, on the cost of maintenance and repairs, are original and conclusive. The argument presented for the lighter loads is so thoroughly practical that further evidence of the same kind is needed to make reply.

The acceptance of these conclusions does not necessarily lead to larger pulleys and more expensive plants, as might at first appear. The burden of the whole argument is for thicker, not wider belts, and such other conditions as will give the greatest economy for the money expended in belting.

Conclusions 27, 28, and 29 are interesting facts concerning the stretch in belting which have not been generally observed or noted by previous writers and experimenters.

Conclusions 30 and 31, on the interruptions to manufacture caused by overloaded belts, are not only of great practical importance, but they suggest another side of the belt question which has never received the attention it deserves.

Conclusion 32, in regard to the best type of belting, is a statement worth noting.

Conclusion 33, in regard to large shifting belts, can be admitted by the writer, but he is not prepared to accept the following conclusion, that shifting belts are much to be preferred to cut-off coupling or friction-clutch pulleys for throwing heavy lines of shafting in and out of use. Under the condition of constant running night and day, experienced by Mr. Taylor, the case may be different; but as far as the economy of belting is concerned there can be no doubt of the value of friction clutches.

The form of belt shifter described in conclusion 35 has been found in the writer's experience to be the only satisfactory method of shifting wide belts.

In reviewing this paper, the most prominent characteristic is the practical information obtained upon new and original lines of investigation.

What little theorizing is indulged in is based upon abundant experience, or upon rigid mathematical deductions.

Possibly the conclusion in regard to maximum speed may be regarded as theoretical rather than practical in character, but this does not affect its truth, which can easily be demonstrated in either way.

An interesting case in point occurred in the writer's experience on a cold-saw, driven by a belt running in the neighborhood of 8,000 feet a minute. No matter how tightly the belt might be laced it would soon stretch and fail to drive at that speed, but by reducing both the driving and driven pulleys in diameter, without altering the width of the belt, it was found that more power could actually be transmitted at a slower speed.

On the whole, the conclusions reached in this paper, although not claimed by the author as finalities, appear to be supported by the strongest kind of evidence, and in the opinion of the writer the thanks of the Society are due to Mr. Taylor and the Midvale Steel Company for the valuable results of their labor and painstaking care.

Mr. Carleton W. Nason.—In reading the author's paper a day or two ago there was one feature in the stretching of the belting which was not very clear to me. He states that under spring-balance tension he draws his clamp so as to give it a pull of 71 lbs. per inch of belt width. Now, if there is very little friction or load on the countershaft, it is quite apparent that the tension will be immediately transmitted to the other side of the pulley, where the belt will also have the same tension.

When the machine is started, then, there will be an additional strain imposed on one side of the belt—viz.: the working load—which will diminish the tension on the slack side a corresponding amount; and it is very likely that under these working conditions, when the belt is first started, on the tight side the tension will exceed, say, 100 lbs. to the inch of belt width.

As he says that, after a given time, the tension on the belt drops to from 25 or 30 lbs., it is, I think, apparent that the percentage of stretch on the tight side is more than is stated. That is, the decrease would be from 100 lbs. on the tight side of the belt, instead of being from 71 lbs., down to the 25 or 30 lbs.

I would therefore like to ask whether the tension given is on the tight side of the belt only, whether it is the average of the

two, or whether it is the tension of the belt when at rest, with both sides under equal stress.

Mr. F. W. Taylor. *—I wish to thank Messrs. Towne, Lewis, Cooper, and Kent for their full and most flattering criticism; and, as the matter of greatest importance, will first refer to the practical suggestion of Messrs. Towne and Kent regarding the next step to be taken in the solution of the belting problem.

I fully agree with them that another shop experiment on a large scale is needed to determine more accurately the most economical "total load" under which belting should run, and would suggest that, if opportunity offers, one-half of a shop should be belted on the basis of an effective pull, for double belting, of 30 lbs. per inch of width, while the belting of the other half should deliver an effective pull of 45 lbs. per inch. I am quite sure that the limit of economy lies between these two points.

Messrs. Cooper and Lewis combine a wider practical and theoretical knowledge of the subject than any two writers with whom I am acquainted, and the criticisms of Messrs. Cartwright and Rogers have been so completely answered by these gentlemen that it would seem superfluous for me to add anything to what they have said.

I wish, however, to alter the impression left on one's mind by reading Messrs. Rogers' and Cartwright's remarks, that the experiment and its conclusions are theoretical vagaries, which, if put into practice, would result in absurdities.

While the experiment was undertaken, to a small extent, in the interest of science, yet its principal and almost sole object was to run the machine shop of the Midvale Steel Company more economically, as far as belting is concerned, than other similar shops were run, and this object has certainly been attained. It has now been more than three years since I left the employ of that company, and although there is probably not a single man at Midvale who cares whether there is a belting experiment going on or not, the method of treating, tightening, and caring for the belts is precisely the same as when I left, and the records are kept just as thoroughly now as then, not in the least in the interest of science, but because it is the

* Author's closure, under the Rules.

cheapest way to care for belting, and the shop earns more money that way.

Mr. Kent in his remarks has given the clearest and best explanation that I have seen of the reason why belts, used apparently under rules differing from one another by as wide a ratio as one to three, will frequently give equal satisfaction and durability.

In answer to Mr. Nason's question as to what is meant by the term "tension" as used in the paper, I would refer him to paragraph 21 in the body of the paper, in which most of the important terms used in the paper are defined.

It is much to be regretted that no uniformity exists in the use of technical belting terms by the various writers on the subject.

In several cases, the conclusions arrived at in the paper, as they are worded, do not accurately express my opinions on the subject, since they are more positive and decided than I feel. This is particularly the case regarding the use of triple and quadruple belts. While up to date my three years' experience in the use of such belts points, almost without exception, to the superiority of thick over thin belts, a period of three years is so small a fraction of the life of a belt that a very imperfect idea as to the relative durability of thick and thin belts can be formed in so short a trial.

During this time I have observed two cases of unusual wear on triple belts, which, if they represent what is likely to be the normal wear of thick belts, would certainly condemn their use.

In these cases the belts began to crack on the inside—that is, the side next to the pulley. In both instances, however, they had been subjected to a large amount of slip, so that the pulleys on which they ran were frequently too hot to be touched with the hand. It is, therefore, not improbable that the ply of leather next to the pulley may have been overheated. Both of these belts, however, ran on pulleys small in diameter, so that it is possible on the other hand that the cracks may have been due to the bending of the belt in rounding the pulleys.

It would seem, therefore, of the utmost importance to know whether thick belts on small pulleys usually give out by cracking on the inside or the outside of the belt, and I shall be greatly indebted to any member of the Society who can enlighten me on this point.

I wish to add a word of warning to those intending to use

narrow and thick belts instead of wide and thin. The pulleys for thick belts should be made heavier in the spokes, as well as the rims, than the standard pulleys made for double belts by the best manufacturers in this country, or trouble will surely follow.

One fact becomes more apparent the further the problems in belting are investigated; namely, that what we do not know about belting far exceeds our knowledge of the subject.

DLXIX.*

RECENT PROGRESS IN THE MANUFACTURE OF STEEL CASTINGS.

BY H. L. GANTT, ELIZABETH, N. J.

(Member of the Society.)

BEFORE beginning the main subject to be treated in this paper, it would seem not inappropriate to point out a few of the steps by which the art of casting steel has arrived at its present advanced state. I must not be misunderstood to mean by this, that the process is not susceptible of improvement beyond what we have already attained; but when we remember what the ordinary steel casting was a few years ago—a rough chunk composed of about equal parts of steel and holes—and note that to-day a steel casting is subjected to a more rigid inspection for cracks and holes than an iron casting, while the general appearance is required to be nearly as good, we are forced to admit that a great advance has been made.

When I spoke of the condition of the art a few years ago, probably but few realized how very few years were really meant. The first soft steel castings, such as are used for gun carriages, that were made in this country, and which were subjected to a ballistic test, were made in 1886 and 1887, by the Midvale Steel Company, who were the pioneers in this work. The ballistic test was considered severe at that time; but to-day the ordinary soft castings made by almost any one of a dozen of the best steel casting companies would pass it successfully.

Though it was about 1886 or 1887 before any large increase took place in the use of steel castings, they had been successfully made in this country at least as early as 1867.† In July of

* Presented at the New York meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† This historical note was furnished some time ago to Mr. Edwin H. Cramp, to go into a paper he purposed to write on the subject of steel castings, but it has seemed to me to be of sufficient interest to publish it here also. If any member has any additional information on this subject, I should be glad if he would add it in the discussion.

that year, the Wm. Butcher Steel Works, now the Midvale Steel Works, cast some crossing frogs for the Philadelphia and Reading Railroad of crucible steel. The moulds were made of ground fire brick, black-lead crucible pots ground fine, and fire clay, and washed with a black-lead wash. The surface of these castings was very smooth, and the metal, which was practically tool steel, was so well adapted for the purpose that some of the castings made in those early days are said to be still in use.

This was before the use of silicon for solidifying steel was known, and the castings were very much honeycombed in all parts except the wearing surface of the frogs, which was "cast down." The possibility of getting one face of a casting approximately solid admitted of the use of steel also for hammer dies, although their general sponginess was a great obstacle to their extensive adoption.

On April 28, 1876, two small hammer dies were made of open hearth steel at the same place, and on May 29th a hammer head weighing 2,535 pounds was made. As long as castings were small, the moulding mixture of ground brick, ground pots, and fire clay was satisfactory; but on large castings made with that facing, the surface was very imperfect, and the sand adhered to it with great tenacity. But little improvement in the method of moulding was made for some time, and while the quality of the steel was gradually improved, the appearance of the castings was decidedly against them, and but little progress was made in their introduction.

The next step taken by the Midvale Steel Company was to leave the ground pots out of the moulding mixture, and to wash the mould with finely ground clay fire brick. This was a great step in advance, especially in heavy castings, and a marked improvement took place in their general appearance; but the mixture still clung so strongly to the casting, that only comparatively simple shapes could be made with certainty. A mould made of such a mixture became almost as hard as fire brick when dried, and offered so much resistance to the proper shrinking of castings that, when at all complicated in shape, they had so great a tendency to crack that their successful manufacture was almost impossible. The next step was to use a mixture of silica sand and flour. In small castings this gave pretty good results, as such a facing cleaned off the casting very readily, leaving a smooth surface much superior in appearance to that

left by the ground brick. The great difficulty in the way of the extended use of such a facing was the tendency of the flour to burn in drying, leaving the sand without a proper bond, and often the burnt mould was so friable as to be entirely useless. In the case of large castings it was almost impossible to dry thoroughly the whole mould without burning that portion nearest the fire, and for such work flour was necessarily abandoned.

By this time the use of silicon for solidifying castings was coming into general use, and the only obstacle in the way of making good steel castings was the want of a suitable moulding sand. This was ultimately found in a mixture, the principal constituents of which were silica sand and molasses thoroughly ground together.

The improvement in the appearance of castings caused by the use of a sand that left a clean, smooth skin was the cause of their rapid adoption; and the year 1887, when the Midvale Steel Company made the first gun carriage castings of steel, is approximately the date at which their rapid extension began.

At this time the necessity for hot metal did not seem to be understood, and castings frequently contained "blow holes," when it seemed that all the conditions for making them perfect had been complied with. It was noted before a great while that small castings contained more holes than large ones, and that those poured last on a heat were worse than those poured first. The conclusions that hot metal was needed to make solid castings, and that smaller castings needed hotter metal than large ones, were soon deduced and amply verified. Apart from these considerations, the superior appearance of castings made from hot metal was sufficient to cause all steel casting makers to work in that direction. The difficulties in the way of getting uniformly hot metal are hard to overcome in an open hearth furnace. The delay caused by a slow or difficult tap of the furnace, which is always liable to occur, is fatal to the production of the best castings. Besides what has already been mentioned, "cold metal" in the ladle is to-day the cause of more complaints by purchasers, and expense to the manufacturer, than anything else in connection with the manufacture of steel castings. I will enumerate a few of the troubles that may be traced directly to this cause. What is known as a "bad shut off," or a leaky stopper, is usually caused by a small amount of

metal chilling in the nozzle, and preventing the stopper from going to its place. In cases where the dribble is slight the worst effect is that a small amount of metal runs into the mould while the ladle is being gotten over the runner. This metal is apt to chill, and make a bad looking spot on the casting, or, in the case of small castings, block up the runner entirely, and cause the loss of the mould. When the leak is very bad, it not infrequently happens, especially when the heat is a large one, composed principally of small castings, that the flasks are covered by the molten steel, and so strongly fastened together that it is almost impossible to separate them from the steel without considerable loss by breakage. Again, by a careful study of the subject, I feel that I have been able to lay to the door of "cold metal" one of the most fertile sources of complaint which steel founders have met with; namely, lack of homogeneity, which includes both "hard spots" and "loose" places. I use the term "loose" rather than "spongy" for the reason that I do not so much mean a cluster of holes as a spot where the metal simply seems less dense. Such places, without any visible continuous hole, or even without anything that would indicate the existence of such a hole, allow leaks in hydraulic cylinders, and cause the failure of a test bar.

When we consider more in detail the process of making open hearth steel castings, we see why these defects are more apt to occur in cold than hot metal. While the metal is being tapped from the furnace, the silico-spiegel necessary to prevent "blow holes" is generally thrown into the ladle, usually in the solid, though heated, state. If the metal is very hot, and consequently thoroughly fluid, the silico-spiegel is dissolved at once, and uniformly disseminated through the mass. If, however, the metal should not be so hot, it is easily seen that the silico-spiegel may not become at all thoroughly diffused, and the slightly viscous metal in the ladle may be the reverse of homogeneous, in which case the resulting castings may be very far from what is hoped for. It is now the practice, in the best foundries, to melt the silico-spiegel, and these defects in their worse form are thus practically eliminated.

Why hard spots should appear in cold metal is readily seen, for lumps of partially melted silico-spiegel would certainly form the nuclei for such spots; but why the "loose" places should occur is not quite so apparent. It may be noted that the loose

places appear to a more marked extent in large masses than in small ones, and the most reasonable explanation seems to be that the silico-spiegel, which is but imperfectly melted when the metal is poured, finally melts thoroughly after the metal is in the mould, which can only occur in very heavy castings, forming a spot higher in carbon, manganese, and silicon than the metal surrounding it. Such a spot would have a lower melting temperature than the adjacent metal, and would remain partially fluid after the other metal was set. The result of such a condition would be to drain away a portion of the fluid metal to feed the shrinkage in the metal solidifying about it, leaving a more or less spongy spot.

That shrinkage holes and cracks do still occur in steel castings is well known, but the success that founders have had in casting complicated shapes within the last year or two has been surprising. The fact that a steel casting *must* cool at the bottom first, and then gradually upward in the mould, is being learned by designers; and shrinkage holes are becoming less frequent, for the founder at once fixes on the heaviest portion for his sink head, which is usually about 20% of the total weight of the casting, and which must remain fluid until the whole of the casting proper is solid.

By a strict observance of the rule that a steel casting should, as far as possible, be of uniform thickness throughout, it has been possible to make, within the past few years, castings twenty feet long by ten feet wide, which would have been considered impossible a few years ago. To one unfamiliar with the details of the process of making steel castings, the statement that to-day we can get large steel castings that are perfectly satisfactory, while it is almost impossible to get good small ones, would seem very strange. It is, however, a fact, and the explanation is simple. Most steel castings are made by the open hearth process, and the furnace usually melts from five to twenty tons. To pour even five tons of steel into castings averaging 100 lbs. in weight would, as a rule, take so long, that those poured last would be of inferior quality on account of the cooling of the metal. As a rule, a large portion of the heat is poured into a comparatively few large castings, and the remainder put into the smaller ones. If the metal is very hot to start with, this may give fair results, but in all cases the least desirable metal is put into the small castings.

This difficulty seems almost insurmountable by the open hearth process, but is readily overcome by a small Bessemer converter, from which we can get two or three tons of metal as hot as we may wish it. Add to this the fact that we can get from the converter that quantity of metal of any composition we may desire, every half hour throughout the day, and we realize the advantage of the Bessemer process for making small and medium weight castings. The manufacturers of electric motors who have been buying their castings from open hearth founderies have apparently not realized what the Bessemer foundry has in store for them. It seems hard to believe that perfectly solid steel castings can be made of the following composition with regularity: Carbon 12%, manganese 30%, silicon 30%, but it is a fact. It may be noted, however, that this steel, while very well adapted to electrical purposes, is of but little value for any miscellaneous castings, except very small ones, on account of the tendency it shows to crack during the process of solidifying, when cast in shapes at all complicated.

The fact that we can make a number of different kinds of steel each day if we use a converter, and but two or three heats at most in twenty-four hours if we use an open hearth furnace, gives a great advantage to the founder using the converter, especially if he is making all kinds of steel castings, enabling him, as a rule, to make far more prompt deliveries than is possible for the open hearth founder doing the same variety of work.

Among the other advances made in casting steel may be mentioned one for making a casting having one or more faces of a steel much harder than the body of the casting. The process consists in lining such faces of the mould as will be adjacent to the parts of the casting it is desired to harden with a metallic alloy in a crushed or powdered state, capable of being melted and absorbed by the molten steel in contact with it, and of such a nature as to impart to the steel a hard face or a face of such a composition as may be readily hardened. If it is desired that the casting shall have a permanently hard face and be used for stamp shoes, crusher jaws, hammer dies, etc., ferro-manganese gives the best results. If it is desired to do machine work on the face, and make it extremely hard afterwards, ferro-chrome is most suitable. The fact that it is possible to produce a soft steel casting, having a face that can be hardened without causing the remainder of the casting to become brittle, will undoubtedly

make it possible to use steel castings in place of chilled iron in many places with great advantage.

DISCUSSION.

Mr. W. F. Durfee.—This paper is probably of more interest to me than it can possibly be to any other gentleman of this Society. It was my lot in 1871 to accept the responsibility of the superintendence of the Midvale Steel Works. I was connected with that institution about three years. The author of the paper, by implication, conveys the idea that though there were some few castings made at the Midvale at about that period, or possibly a little earlier, that the castings were not numerous, and though they were fairly good, the industry was not of very much importance. At least I think that would be the impression that a casual reader of the paper would derive from it. This is far from being a correct impression. At that period the Midvale Steel Works had on their regular sales list 31 patterns of reversible frogs. There was one road that had over 600 of those frogs in use. Another road had over 400 in use, and a great number of other roads around the country had various smaller numbers. During my connection with the works I never knew of one of those frogs being returned. When they wore out on the upper surface they were turned bottom side up. These frogs varied from about 250 lbs. in weight up to about 900 lbs. in weight. They were from about 30 inches to about 6 feet in length. Most of them had three lugs on each side, which projected from the body of the frog and were not particularly pleasant things to have, in the matter of shrinkage in the casting. But there were very few frogs lost on that account.

Then in regard to the soundness of small castings, he speaks as though there were some recent improvements in that line. The Midvale Steel Works at that time made sound castings, and they made castings that were not sound. I think that is true of every steel-casting concern to-day. I do not believe it was any more true at that time than at present. Since leaving that institution I have had occasion to use a great many steel castings from different manufacturers, and I have never found an instance where any manufacturer uniformly made sound steel castings from the same patterns.

As regards the alleged unsoundness of small castings at that

time, I recall one illustration of the soundness of them, and the practical uniformity of soundness so far as the strength of castings was concerned, and the uses to which they were applied. At that time we made a number of "trestle trees" for the topmasts of ships. It was a patented idea of some person whose name I do not recollect. They were very light, and had two horns or projections on each side. We never had any of those returned to us. But the inventor came to me one day and said: "If I could show the captains and owners of ships that these things were strong, I could get a great many more orders." I said: "I will tell you what to do. You bring the upper section of a topmast here, and I will shove one of those trestle trees onto it until the wood is crushed." We had a hydraulic press with which we pulled apart 6-inch steel bolts occasionally, very much to our disgust, and I said: "I will put that topmast into our press and shove that trestle tree on until it crushes the wood, and no service to which it could be applied will be more severe than that." My suggestion was carried out, and the orders for trestle trees increased materially in consequence.

With regard to smoothness of steel castings made at that time, we had no trouble in turning out smooth castings of large weights and small weights. I say large weights. We made pretty heavy pieces—an anvil block. I do not remember what it weighed, but it was some 2½ feet high and perhaps 2 feet square at the bottom, and it was cast with a horn on one side. After a while those horns, under the severe blows of the hammer, would break off, and it was expensive to replace those things with new castings. I introduced a plan there, of having a diagonal hole bored through them and putting in forgings for the horn. The rolls for the tire mill were of cast-steel in my time, and we had no trouble with holes in them.

As to the matter of facings, we had no difficulty in getting smooth steel castings with a facing composed of the materials that the gentleman describes in the paper. But there are a great many ways of using a given kind of material, and perhaps the subsequent ways of using those materials were not as good as they were in my time. I do not know anything about that. I do know, while I was there, that we cast bevelled gears and spur gears of various sizes, and some of them pretty large, and we cast a great many anvils. The implication here is that there were only two or three small anvils cast at that time. We

made a great many anvils at that period, not only for ourselves but for others.

There is another thing not mentioned in the paper, and that is that at that time all the steel castings were made from crucible steel. There were none made from open-hearth or Bessemer. The suggestion made of the use of the Bessemer converter in connection with the open-hearth furnace was a very good one.

As to the discussion of the advantage of hot metal for making steel castings, that is spoken of as being a comparatively recent discovery. I do not know of the time when it was not self-evident to me that hot metal was desirable for making sound castings. I presume that that is and always has been a part of the common knowledge of all people who make castings in iron and steel, and I have always supposed that it always had been the common knowledge that hot metal was desirable to produce the best results. I do not regard that as anything new, and, as I say, I do not see anything in the paper that was not known at that time, when the Midvale Steel Works started, except this matter of the use of the Bessemer converter in connection with the open-hearth furnace.

Mr. Fred. W. Taylor.—I represent a rather half-way position between Mr. Gantt and Mr. Durfee at the Midvale Steel Works, having come there about the year 1877, and left in 1890. I think I can to a certain extent reconcile the differences between the two gentlemen. As Mr. Durfee says, the castings made in his time were made entirely by the crucible process, whereas those in Mr. Gantt's time were made almost entirely by the open-hearth process. A few crucible castings are still made at Midvale, but crucible steel is now chiefly used for filling up risers or sinking heads in large castings, as the shrinkage takes place in cooling. Now the largest difference between the castings of Mr. Durfee's day and those of the present time lies in the quality of the metal—that is, in the amount of carbon contained in the metal. The old crucible castings used to be almost universally made of metal high in carbon relatively to the metal from which castings are now made. That is to say, modern steel castings are largely used to replace wrought-iron forgings. Therefore they are made of a comparatively low carbon, while the earlier castings were made of steel more nearly of the composition of tool or spring steel, so that when you

speak of hot metal in castings it is entirely a relative term. A hot metal for a very low-carbon steel is a decidedly different matter from a hot metal for a high-carbon steel. That is, if you have a metal which is sufficiently hot to pour perfectly fluid with a high carbon, it will be a cold metal for a low-carbon steel.

I remember distinctly the different classes of moulds which were used at Midvale from the time I came there, when Mr. Durfee's moulds, consisting of ground pots and bricks, were still in use, and the stages we passed through in getting to the present mixture of as nearly as possible pure ground quartz cemented with molasses. While what Mr. Durfee says is entirely so, that in a great many cases the old ground-pot mixture would clean quite freely and make a good surface, yet when we came to make large castings where we used low-carbon hot metal, which has a temperature enormously greater than the hot metal of high carbon, the moulds would in many cases burn on to the metal, or the metal would penetrate the mould in a spongy mass, so as to require days of chipping. I have no doubt that Mr. Durfee's chipping gang, after anvil blocks and gear wheels were cast, used to have days of chipping in removing the sand from between teeth and from various parts of the surface. At any rate, while using the same mixture as that used in Mr. Durfee's time, we had a chipping gang on hand throughout the year to chip this mixture of part steel and part moulding sand from the surface of the castings. When we found the surface it would frequently be very good and very sound; but steel castings were not a commercial product as far as making money went at that time. We lost money on steel castings regularly for years while we were stepping from one mixture to another, until the mixture at present in use was reached. This mixture, if properly used, cleans for small castings in many cases as easily as the ordinary cast-iron moulding mixtures. Of course, if an accident occurs, or improper moulding occurs, so that the steel runs into the sand of the mould, you still have the same trouble with cleaning a spongy surface from the casting that there used to be. But, generally speaking now, the art of cleaning steel castings is a very simple one as compared with what it was in the early days, and I think that has a great deal to do with the fact that the manufacture has now become a commercial success.

Mr. Durfee.—There is one point I intended to speak of, and that was in regard to the knowledge of the use of silicon for

making sand castings in this country. I do not know to what extent silicon was employed in making steel castings, but there was no reason why silicon should not have been employed by makers of steel castings, because the knowledge of it was promulgated as early as 1861 and 1862. In one of Bessemer's patents dated February 1, 1861, he claims distinctly the use of silicon for the prevention of blow-holes.

One word in regard to the criticism in relation to making money in steel castings at that time, and the roughness of gears and things that were cast. I have no recollection of any unusual or extraordinary amount of chipping on the steel gears cast at that period, except in one particular—it was generally quite difficult to get off the "sprues." As regards the spaces between the teeth, and everything of that sort, they cleaned out nearly as well as the average cast-iron castings. I had no difficulty of the kind, and we kept no permanent chipping gang. The balance sheet at the end of the year showed a profit on steel castings. I have heard a rumor of a great deal of trouble that followed my time. I do not know anything about it myself, and Mr. Taylor's remarks furnish the first authentic information I have received in regard to that matter. I have no doubt whatever that Mr. Taylor has told the truth in regard to the state of facts that existed at that period. We did not in my time make steel castings of one temper. Some of the castings were hard, and made hard on purpose; some were soft, and made soft on purpose. I think we cast for the first "blooming mill" made in this country the bearing blocks for the middle roll, a couple of vertical holes bored through them in which screw threads cut for the vertical screws. Any one who recalls the original construction of blooming mills will remember, perhaps, the feature that I speak of. We did not make absolutely sound castings all the time. But I do not believe that Midvale, or any other steel works of the present time, does any better in that regard than was done at Midvale in the years 1871, 1872, and 1873.

Mr. John Platt.—As long ago as fifteen or sixteen years a firm in England used large numbers of complicated steel castings for hydraulic machinery, and as many of these had to stand high pressures they had to be very sound. More or less trouble was, of course, had with them, and the experience obtained leads one to some of the conclusions arrived at by Mr. Gantt in his paper.

He says that large and small castings cannot be poured from the same heat with any success, and this we also found, as it was hardly possible to obtain both large and small castings that were sound from the same firm.

The paper and discussion would give one the idea that it is only during the last two or three years that sound castings have been able to be turned out with any degree of certainty.

The castings before referred to were made by Messrs. Hadfield, of Sheffield. They have for fifteen years been of a surprisingly uniform character. We hardly ever had a defective one. They devoted their attention to one class of work, and gave care to their mixtures. Another firm made, at a much lower price, very large castings, and with the same success, but this latter firm could do no good with small ones. Trying to do too much in one foundry has been responsible for a great deal of trouble.

Again, with regard to uniformity of design, and letting the metal be as equal in proportion as possible—that has a great deal to do with success or otherwise. I have seen castings made with small fillets that have broken in a short time, which with large fillets and a better disposal of the metal would stand perfectly well. This is particularly the case under heavy and repeated loads, constantly varying from nothing to a maximum. I remember a case in which a casting was machined out with a sharp corner for some experiments. The stress was as low as four tons per square inch, but with a few weeks' work the casting broke. A small crack started up in the corner, and the constant change from no load to full load soon caused a fracture.

Mr. W. A. Pearson.—The point in steel castings which designers should consider very carefully is to get the piping in the centre of the walls of each member of the castings. The only way that this can be accomplished is by putting in fillets large enough, so that the line of expansion and contraction is as near uniform as possible upon both walls of casting in relation to the centre line of casting. If it is a cylinder with ribs on it the fillets should be large enough, and the outer corners should conform as near as possible to the circle of the fillet. All ribs on arms or rims of wheels, or at any point of increase or decrease in the casting, should be so designed that there is as uniform a strain on the walls of the casting as possible. In this way the piping will be in the centre of the two walls of

casting. This point will also aid in getting rid of the gas in the metal and mould, which is another important factor to be considered to insure good steel castings.

The above principles regarding steel castings will apply equally as well to iron, and if carried out the result will be stronger castings.

Piping exists to a much greater extent in steel castings, as the contraction is about three times as great as in iron castings. The walls solidify on the exterior as the casting cools off, thus causing a strain from the centre of walls. If all designers of steel castings will go thoroughly into the points in question, they will improve the quality of steel castings.

Mr. Harrison A. Royce.—I would ask the attention of the Society for a moment to one of the practical questions which arise from this very interesting discussion, and that is this:

It is granted that defects exist in steel castings; also that they are found in larger proportion in small castings than in large ones. Now, is there no practical way by which we can repair those castings so as to make them in every sense commercial?

In my connection with the Thomson Electric Welding Company I have for the last two years given personal attention to this specific matter, and the results of our experiments are to us very interesting, and may be so to the members here.

We have found that by using the arc welding process, and by springing an arc over a blow-hole in the casting, we are able in a very short time to reduce steel filings or steel fragments placed in the cavities, to a fusion, so filling up these holes and rendering the casting homogeneous.

Our first experiments were on a casting of, say, 50 lbs. At first we took the casting cold, as it came to us from the foundry, but we then found that after we had succeeded in filling the holes, that the chilling surface of the surrounding metal prevented a perfect union, causing frequently something like a "cold shut," and therefore was not successful. This led us to try the following method:

The casting was, say, 100 lbs. We first placed it in an oil or gas furnace, and brought the mass to a dull red heat; then springing the arc over the blow-hole, using the metal itself as a positive pole and the carbon as a negative pole, we were able to bring the surrounding surface in about two or three minutes to a proper degree of heat; and by pouring into the cavity steel

filings or small fragments of steel which had been previously treated, we could quickly reduce them to a state of fusion, thus completely filling the holes with molten metal, which, on cooling, proved to be thoroughly united with the rest of the mass, and not showing any defects in hardness or in quality from the rest of the casting.

We have done this in many instances, and find it to be perfectly successful.

This process can be applied commercially, and we find, furthermore, that for an expense of two or three dollars for labor, etc., we are able to save castings which are of much value.

We know from our own experience that imperfections exist; we find such imperfection in our dynamo frames and other castings which we have received from our foundry; we are obliged to reject them, and they probably go into scrap. It has become perfectly practical, by the method of which I speak, to remedy these defects, and this process can easily be made available for all producers of steel castings.

In Russia this process is carried on to a considerable extent, while in England it has become very successful, the firm of Lloyd & Lloyd using it constantly in their business.

Mr. John Fritz.—I had not proposed to enter into this discussion, but there has been a good deal said about steel castings, and by people who seem to know a great deal about them. I would like to ask whether anybody can give any information about the action of basic open-hearth steel in castings; how would it work, and would it be better than a casting of acid steel? I want to find out whether any one present has had any experience with it.

This discussion, too, has taken rather a serious turn, and I am tempted to throw a little fun into it. A friend of mine, who probably dates back prior to 1876, which has been a date mentioned, had a great deal of trouble in cleaning his castings; he was so troubled that he did not know what to do with them. One day he had a specially difficult one to make, and he made a good acid bath, put his finished casting into it, and let it lie there two or three days. He said at the end of that time he found the sand all right, but he couldn't find the casting.

Mr. Thomas R. Almond.—Mr. Platt made mention of Mr. Hadfield. I think Mr. Hadfield made his first experiments near

Boston in 1874. Can you say anything about that, Mr. Platt?

Mr. Platt.—No; I do not know where he made his first experiments.

Mr. Almond.—At that time I was in Winchester and had occasion to use some steel castings. Mr. Hadfield made them for me, weighing only two pounds each, and they were perfectly clean castings. They came out of the sand as readily as ordinary cast-iron. I mention that because the matter of date seems to be interesting. This was in 1874.

Mr. C. W. Baker.—I presume most of the members here saw that very remarkable steel casting exhibited by Messrs. Krupp at the Chicago Exposition, consisting of a cast-steel locomotive frame for a standard Pennsylvania Railroad consolidation locomotive. I would like to ask Mr. Gantt whether it would be possible for any steel foundry in the United States to reproduce that casting, and if not, whether it is merely a matter of size of plant, or a difference in the process pursued in the foundry that would make it impossible.

Mr. Gantt.—In reply to that question there are several points to be noted. First, that casting was machined all over, and nobody knows exactly how much metal was taken off in machining. I think there are perhaps three or four foundries in this country which could make such a casting. They would perhaps lose two or three castings before they got it, but I firmly believe it can be produced in this country.

Mr. Baker.—But you do think it could be produced commercially?

Mr. Gantt.—Certainly.

Mr. Baker.—As a commercial success in competition with forged frames?

*Mr. Gantt.**—Certainly, after they got into the manufacture. The first one would probably not be a commercial success, but an order for a hundred would, and after that undoubtedly they would be made with uniform success. Steel castings of complicated shape are difficult to make, and each one is more or less an experiment. You have to go at such work with that point in view.

I was very glad to hear Mr. Durfee's remarks, because this

* Author's closure, under the Rules.

paper was designed largely to bring out those facts that he has stated. We want to find out exactly what has been done in this country in the way of making steel castings, and find out who is to have the credit for it. I have seen a number of the castings of which Mr. Durfee spoke, some of which are in use still. They were made of crucible steel, which is usually more solid than open-hearth steel, and probably they had fewer blow-holes in them than the ordinary open-hearth castings made at that period. I have never seen any, however, that I considered sufficiently perfect to send to a first-class machine shop.

With reference to the cleaning of castings, it just happened to be my luck to find when I went to the Midvale Steel Works some people in the casting department who had bridged over the whole gap from the time Mr. Durfee was there until the time I went there. The facts that I put in my paper I obtained from records which I found in the possession of the company, and the general impressions came from the men who made the castings. When, some years ago, we made a large hammer die, from which the sand fell off, leaving a perfectly smooth casting, the man who had charge of the stripping came to me in great delight, and said : "The first casting we made like that, it took a week to find." That was all. There is also on record at the Midvale works a piece-work rate of $2 per tooth for cleaning some roll pinions made for the Scranton Steel Works. I do not mean to say that some people did not make castings that cleaned more easily.

Mr. Durfee.—One moment. Lest your remark in regard to the $2 per tooth should convey a false idea, I should say that the Scranton Steel Works was built long after I left the Midvale Steel Works.

Mr. Gantt.—I do not mean to say at all that I am criticising Mr. Durfee's remarks. When I speak of heavy castings, I do not mean castings weighing 500 lbs.; I mean castings weighing 10,000 lbs. Possibly that may clear up some things. These pinions weighed 13,000 or 14,000 lbs., and they were about six inch pitch, herring-bone teeth. While it took a couple of weeks to clean them at that time, they are cleaned now in a day and a half. The reason why I make the suggestion that a mixture of ground clay, ground black-lead pots, and fire brick was difficult to clean on a large casting of low carbon, was that the intensely hot metal absorbed the carbon from the moulding mix-

ture while it was cooling, and changed the composition to quite a considerable extent. The remainder that was left after the carbon was absorbed adhered to the casting with great tenacity.

As to the use of silicon for solidifying castings, a royalty was paid to Alexander Lyman Holley's estate until about 1890 for the use of silicon for that purpose. It was perhaps only for the special way of introducing it, but such a royalty was paid, indicating that such use of silicon did not begin until 1873.

DLXX.*

A COMPARISON OF THE MEAN EFFECTIVE PRESSURES OF SIMULTANEOUS CARDS TAKEN BY DIFFERENT INDICATORS.

BY D. S. JACOBUS, HOBOKEN, N. J.

(Member of the Society.)

THE conclusions derived from the data obtained by the tests are:

1. With correctly fitted pistons in the indicators there need be no greater error involved in their use than that which occurs in measuring the hot scales of their springs.

2. A leaky piston is much more reliable than one that is too tight a fit. A piston which will not fall through the cylinder by its own weight may be too tight and produce errors due to undue friction. The friction of an indicator piston should be so small that there will not be over the thickness of a very fine pencil line between the lines obtained for a rising and falling steam pressure in the tests to determine the hot scale of the springs.

3. The greatest difference found between two indicators with correctly fitted pistons for the mean effective pressures of cards taken at one-quarter cut-off was, as a maximum, 0.9% in four indicators tested, and the greatest difference in the measurement of the mean height of the diagrams for any two indicators was .005 inch. This was for comparative tests in which the average of either 12 or 24 cards was taken. The greatest difference between any single pair of indicator cards taken simultaneously was 2.5% of the mean effective pressure, and the corresponding difference in reading the mean height .017 inch.

4. The greatest difference found between any two indicators with correctly fitted pistons for the mean effective pressures of cards having one-fortieth cut-off, so that a loop was formed below the atmosphere line, was .6 lb. per square inch, and the greatest difference in the measurement of the mean height

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was .010 inch; or, adding opposite signs, .017 inch. The presence of a loop causes the effect of friction and lost motion to be a maximum, and the above differences are therefore greater than were found in cards having no loop.

5. The variation of the weight of the moving parts within the limits now employed by the makers of the standard instruments does not affect the accuracy of the results obtained by the indicators.

Four makes of indicators were tested; these are marked in the tests Old Form of A, New Form of A, B, and C. The tests were made by taking simultaneous cards from one end of the cylinder of a 7-inch by 14-inch horizontal engine running at from 175 to 200 revolutions per minute. The steam pressure averaged about 95 lbs. per square inch above atmosphere. The indicators were attached at the two ends of a tee connected to the end of the cylinder, and were reversed in the middle of each test. After every three cards the indicator pistons were removed and the cylinders and pistons cleaned and oiled. The strings driving the indicator drums were made as short as possible, and wire cord was used to connect them with the parallel motion.

The hot scales were determined as follows: The indicators were subjected to a steam pressure which was measured by a gauge so arranged that it was perfectly cool. The pressure was adjusted to the desired amount, and lines were traced by the indicator pencil. In making a record the pencil of the indicator was first raised slowly, and gradually released, and a line traced; it was then pressed downward, and released gradually, and a second line traced. For indicators in correct adjustment these lines coincided exactly, or so nearly that there was not the thickness of a very fine pencil line between them. The scales derived by the above method were found to agree with those made in tests at the Brooklyn Navy Yard, in which the indicators were compared directly with a mercury column.

Tables I., II.$_a$, and II.$_b$ give the final results of the tests.

TABLE I.

Comparison of Indicators by Tests at ¼ Cut-off.

Corrections made for all variations in the hot scales of the springs.

No. of comparison.	No. of test.	Indicator.	Average mean height of card.	Equivalent scale of spring.	Mean effective pressure in lbs. per sq. inch.	Difference in Mean Effective Pressures Referred to Indicator old A.		
						In lbs. per sq. inch.	In per cent. of total M. E. P.	Equivalent in height of card in inches.
1	1	Old Form of A.	.778	58.3	45.4	.....	.....	.....
		New Form of A.	.556	80.9	45.0	−0.4	−0.9	.005
2	2	Old Form of A.	.675	58.3	39.4	.....	.....	.....
		B.	.675	58.2	39.3	−0.1	−0.3	.002
3	3	Old Form of A.	.610	58.3	35 6	.....	.....	.....
		C.	.738	48.6	35.9	+0.3	+0.8	.005
4	4	Old Form of A.	.698	58.3	40.7	.....	.....	.....
		B.	.700	58.2	40.7	0	0	0
5	6	Old Form of A.	.684	58.3	39.9	.....	.....	.....
		B.	.687	58.2	40.0	+0.1	+0.3	.002
6	5	Old A with weighted piston.	.697	58.3	40.6	.....	.....	.....
		B.	.697	58.2	40.6	0	0	0
7	7	Old A with loose piston.	.681	58.9	40.1	.....	.....	.....
		B.	.691	58.2	40.2	+0.1	+0.2	.002
8	8	Old A with tight piston.	.673	58.3	39.2	.....	.....	.....
		B.	.678	58.2	39.4	+0.2	+0.5	.003

In obtaining the equivalent scales of the springs, the cards were divided up into a number of small areas, and the correct scale was applied to each area. For details of the calculations see Appendix.

TABLE IIa.

Comparison of Indicators by Tests at ¼ Cut-off.

Hot scales of springs determined at about one half the initial pressure.

Number of comparison.	Date of test.	Number of test.	Number of table giving data in detail.	Indicator used.	Steam pressure in lbs. per sq. in. above atmosphere.	Revolutions per minute.	Weight of moving parts of indicators, including gravity component of pencil motion.	Hot scale of spring.	Mean effective pressure in lbs. per sq. in.	Difference in mean effective pressures referred to old A.			Greatest difference in M. E. P. between any two simultaneous cards.	
										In lbs. per sq. in.	In per cent. of total M. E. P.	Equivalent in height of card in inches.	In lbs. per sq. in.	In per cent. of total M. E. P.
1	Oct. 16.	1	V.	Old Form of A.	104.2	191.3	18.5	58.8	45.7	.....	.....	.....	.....	.....
				New Form of A.	104.2	191.3	24.8	81.7	45.4	−0.3	0.7	.004	−1.0	2.2
2	Oct. 16.	2	VI.	Old Form of A.	94.9	190.8	18.5	58.8	39.7	.....	.....	.....	.....	.....
				B.	94.9	190.8	30.5	58.6	39.6	−0.1	0.3	.002	−1.0	2.5
3	Oct. 16.	3	VII.	Old Form of A.	90.7	194.8	18.5	58.8	35.9	.....	.....	.....	.....	.....
				C.	90.7	194.8	27.0	48.8	36.0	+0.1	0.3	.002	+0.8	2.2
4	Oct. 20.	4	VIII.	Old Form of A.	95.0	190.7	18.5	58.8	41.0	.....	.....	.....	.....	.....
				B.	95.0	190.7	30.5	58.6	41.0	0	0	0	−0.3	0.7
5	Oct. 24.	6	X.	Old Form of A.	94.3	196.3	18.5	58.8	40.2	.....	.....	.....	.....	.....
				B.	94.3	196.3	30.5	58.6	40.2	0	0	0	+0.4	1.0
6	Oct. 20.	5	IX.	Old A with weighted piston.	94.2	192.1	37.0	58.8	41.0	.....	.....	.....	.....	.....
				B.	94.2	192.1	30.5	58.6	40.8	−0.2	0.5	.003	−0.6	1.5
7	Oct. 24.	7	XI.	Old A with loose piston.	94.5	192.9	18.9	59.9	40.8	.....	.....	.....	.....	.....
				B.	94.5	192.9	30.5	58.6	40.5	−0.3	0.7	.005	−0.7	1.7
8	Oct. 24.	8	XII.	Old A with tight piston.	94.2	192.5	19.2	58.8	39.6	.....	.....	.....	.....	.....
				B.	94.2	192.5	30.5	58.6	39.7	+0.1	0.3	.002	+0.7	1.8

TABLE II*b*.

Comparison of Indicators by Tests at $\frac{1}{40}$ Cut-off.

Hot scales of springs determined at about one half the initial pressure.

Number of comparison.	Date of test.	Number of test.	Number of table giving data in detail.	Indicators used.	Steam pressure in lbs. per sq. in. above atmosphere.	Revolutions per minute.	Weight of moving parts of indicator, including gravity component of pencil motion.	Hot scale of spring.	Mean effective pressure in lbs. per sq. in.	Difference in mean effective pressures referred to Old A.		Greatest differences in M. E. P. between any two simultaneous cards.
										Actual in lbs. per sq. in.	Equivalent in height of card in inches.	
9	Oct. 27.	9	XIII.	Old Form of A.	94.3	180.7	18.5	58.8	7.2	.....	.....	.....
				B.	94.3	180.7	30.5	58.6	6.6	−0.6	.010	−0.8
10	Oct. 27.	13	XVII.	Old Form of A.	94.3	174.2	18.5	58.8	7.1	.....	.....	.....
				C.	94.3	174.2	27.0	48.8	6.9	−0.2	.004	−0.6
11	Oct. 27.	14	XVIII.	Old Form of A.	94.8	176.0	18.5	58.8	6.9	.....	.....	.....
				New Form of A.	94.8	176.0	24.8	81.7	7.4	+0.5	.006	+0.9
12	Oct. 27.	10	XIV.	Old A with weighted piston.	94.8	176.7	37.0	58.8	7.1	.....	.....	.....
				B.	94.8	176.7	30.5	58.6	6.4	−0.7	.012	−0.9
13	Oct. 27.	12	XVI.	Old A with loose piston.	90.7	169.8	18.9	59.9	6.7	.....	.....	.....
				B.	90.7	169.8	30.5	58.6	6.5	−0.2	.003	−0.5
14	Oct. 27.	11	XV.	Old A with tight piston.	93.8	175.5	19.2	58.8	8.0	.....	.....	.....
				B.	93.8	175.5	30.5	58.6	6.6	−1.4	.024	−2.0

In calculating the mean effective pressures in Tables II.$_a$ and II.$_b$ the hot scales of the springs employed were those determined over a range of pressure equal to about one-half the initial pressure. In Table I. the scales employed were those corresponding to the height on each strip into which the diagram was divided, so that corrections are made for all variations in the scales of the springs. The hot scales at various pressures, together with the method for allowing for the same, are given in the Appendix.

An examination of Table I., in which corrections are made for all variations of the scales, shows that the greatest difference found between the mean effective pressures for any two indicators was about one per cent., which is equivalent to a height measured on the diagram of 0.005 inch. This height, however, represents the error of standardizing the indicator springs, because the heights in testing the springs were not measured closer than this amount. It therefore appears that there is no difference shown in the results in Table I., where corrections are made for all variation of scales that may not be caused by errors in standardizing the springs.

The greatest difference between the mean effective pressures for correctly fitted pistons, with scales determined at about one-half the initial pressure, as given in Table II.$_a$, and the results in which corrections are made for all variation of the scales, as given in Table I., is 0.04 lb. per square inch, which is equivalent to 0.007 inch measured on the diagram, and equal to about one per cent. of the mean effective pressure.

If the scales had been determined at the initial pressure of 80 lbs., the greatest difference between the results and those in which corrections are made for all the variations of the scales would be 0.3 lb. per square inch, or about eight-tenths of one per cent. The greatest difference between the results for two indicators, obtained by adding differences with opposite signs, would, however, be about two per cent., or twice that found when the scales employed are those determined at about one-half the initial pressure.

If the scales were those determined at 100 lbs. pressure, the results would differ by about three per cent. as a maximum from those in which the scales are corrected for variation, and, adding opposite signs, discrepancies of five per cent. would appear. These great differences are caused by the variation of the scales

of the springs at their maximum limits, and show that the scales for the mean pressures are more reliable than those determined at the extremes of pressure.

Tables I. and $II._a$ contain the results of tests at one-quarter cut-off. Comparisons Nos. 1 to 5 are for indicators with ordinarily fitted pistons that were in proper adjustment, as shown in the tests to determine the hot scales.

In comparison No. 6 a weight was attached to the piston of the Old Form of A indicator equal to the weight of the moving parts, including the gravity component of the pencil motion; the addition of this weight, which made the mechanism heavier than that in any of the current makes of indicators, did not introduce an error.

Comparison No. 7 gives the result of a tight piston in the Old Form of A indicator. This piston would fall through the cylinder of the indicator by its own weight when all oil was removed, but in the tests to determine the hot scale the distance between the lines, on pressing the pencil up and down, was about one-sixty-fourth of an inch. The results show that the mean effective pressure is the same as for the correctly fitted piston. This is probably due to the fact that the form of the card is such that the friction tends to produce too high a back-pressure line, as well as too high an expansion line, and the error is thus neutralized. It was to introduce the error of friction so as to make it conspire to add to the area of the card that the tests at the short cut-off given in Table $II._b$ were projected. The cards taken in making these comparisons gave a loop in the expansion line so that any friction or lost motion tended to add to the area of the diagram. An examination of this table will show that the discrepancies in the mean heights of the cards are about the same as the cards at one-quarter cut-off for all the comparisons except No. 14, which is for the tight piston. The effect of the tight piston can be seen by examining the loop on cards Figs. 80 and 83, one of which is taken with the ordinarily fitted piston and the other with the tight piston.

TABLE IIIa.

Determination of Hot Scales of Indicator Springs. 51 lbs. Pressure per Square Inch

Tests Made October 17, 1893.								Tests with Three Sizes of Pistons in Indicator Old A.			
Old Form of A Indicator.		New Form of A Indicator.		Indicator B.		Indicator C.		Date of test.	Tight Piston.	Loose Piston.	Ordinary Piston.
Card.	Height in inches.	Card.	Height in inches.	Card.	Height in inches.	Card.	Height in inches.				
A.	.870	G.	.625	A.	.875	M.	1.040	Oct. 23.	.865	.855	.865
F.	.865	H.	.625	B.	.875	N.	1.045	Oct. 23.	.870	.850	.865
G.	.865	I.	.625	C.	.870	O.	1.050	Oct. 23.	.865	.850	.865
L.	.865	J.	.625	D.	.870	P.	1.045	Oct. 23.	.860	.855	.865
M.	.865	K.	.620	E.	.870	Q.	1.050	Oct. 24.	.860	.850	.870
R.	.875	L.	.625	F.	.865	R.	1.045	Oct. 24.	.860	.845	.865
								Oct. 24.	.870	.855	.870
								Oct. 24.	.870	.850	.870
Average..	.868		.624		.871		1.046		.865	.851	.867
Scale.....	58.8		81.7		58.6		48.8			59.9	

The determinations of the hot scales for the three sizes of pistons for the Old Form of A Indicator were made both before and after the dynamic tests comparing the pistons. As the tight and the ordinary fitted piston gave heights in the tests for the hot scale which agreed within the errors of observation with the values determined on October 17, the scale obtained on October 17 was adhered to in calculating the mean effective pressures from the Indicator Cards.

TABLE III*b*.

DETERMINATION OF HOT SCALES OF INDICATOR SPRINGS.

Tests to show that the springs did not change during the time that they were in use. 51 lbs. pressure per square inch.

OLD FORM OF A INDICATOR.		NEW FORM OF A INDICATOR.		INDICATOR C.	
Date.	Height in inches.	Date.	Height in inches.	Date.	Height in inches.
Nov. 8.	.870	Nov. 8.	.620	Nov. 8.	1.045
Nov. 8.	.870	Nov. 8.	.620	Nov. 8.	1.045
Nov. 8.	.865	Nov. 8.	.620	Nov. 8.	1.050
Nov. 8.	.870	Nov. 9.	.630		
Nov. 9.	.870	Nov. 9.	.630		
Nov. 9.	.870	Nov. 9.	.630		
Nov. 9.	.870				
Average	.869		.625		1.047

The scales of Old and New A and of C are shown by the above to have remained unaltered during the tests. The scale of Indicator B could not be determined at the time the above tests were made, because the cylinder had been removed from it and made to fit over a special tight piston. The comparative tests of Old A and Indicator B in ordinary condition that were made before each special test show, however, that there was no change in the scale of Indicator B.

Tables III.$_a$ and III.$_b$ give the readings obtained in determining the hot scales of the indicator springs. The basis of pressure was the Utica Steam Gauge Company's device, in which a flat plate is held down by a weight over a one-inch-square opening having knife edges; a fluid pressure of oil is made to act under the plate, which will lift when the pressure in pounds per square inch is equal to the weight holding down the plate. The hot scales were redetermined at intervals during and at the close of the dynamic tests to make certain that there was no change.

Table IV.$_a$ gives the results of tests made at the Brooklyn Navy Yard, through the courtesy of Commodore George W. Melville, together with those made by ourselves at the same pressures; and Table IV.$_b$ gives the observations in detail for one set of the Navy Yard tests. The tests at the Navy Yard were made after all our tests had been completed.

TABLE IV*a*.

COMPARISON OF SCALES OBTAINED IN TESTS AT THE NAVY YARD AND BY EMPLOYING OUR OWN STANDARD.

Indicator.	Pressure in lbs. per sq. inch above atmosphere.	Scale of Diagram in lbs. per sq. inch.		Corresponding difference in measuring diagram in inches.
		Navy Yard mercury column.	Derived from gauge standardized by weight device.	
Old A	25	57.1	56 8	−.002
Old A	50	58.7	58.8	+.002
New A	25	78.9	79.4	− 002
New A	50	81.4	81.7	+.002
New A	100	80.2	80.1	−.001
C	25	47.9	47.9	.000
C	50	48.7	48.8	+.002
C	100	50.0	50.0	.000

The greatest difference between the average results of the tests made at the Navy Yard and those made by ourselves is equivalent to a height on the diagram of .002 inch, which is within the error of duplicate measurements.

A gauge was compared with the Utica Company's weight device and carried to the Navy Yard, together with the indicators. The corrected readings of the gauge agreed with the Navy Yard standard.

METHOD OF MEASURING THE MEAN EFFECTIVE PRESSURES.

The indicator cards were measured by two parties. The first observer measured the area by going around each card twice with an Amsler planimeter, and took a mean of the two readings, provided they agreed to within .02 square inch. If the discrepancy was greater than this, the card was gone over until it fell within the above amount. The results were then tabulated. The second observer worked in a similar way and prepared a second table, which was compared with the one obtained by the first observer. In general the reading in these tables agreed to within one or two hundredths of a square inch, and, if this was the case, the mean value was employed in determining the mean height. In a few cases, however, mistakes in reading gave larger discrepancies, which were corrected before obtaining the mean values.

TABLE IV*b*.

TESTS OF INDICATORS AT BROOKLYN NAVY YARD.

Measurements in detail for a given pressure.

Heights in inches corresponding to 50 lbs. per square inch above the atmosphere.

Number of Card.	Indicator Old A.	Indicator New A.	Indicator C.
1	.855	.615	1.030
2	.850	.615	1.030
3	.850	.615	1.020
4	.855	.610	1.030
Average	.852	.614	1.027
Scale	58.7	81.4	48.7
Heights in inches, corresponding to scales determined by the Utica Co.'s weight device	.850	.612	1.025
Difference in height on diagram in inches, between our own and the Navy Yard tests	.002	.002	.002

The planimeters were tested by revolving the tracing point in a circle by means of a radius piece. Each instrument was tested over several sizes of circles. The readings for each revolution were taken for ten consecutive revolutions, and the differences of these readings were found to agree within .03 square inch for an area larger than any of the indicator cards, which indicated that the instrument would give concordant readings to within .01 of a square inch. This is a severe test for a planimeter, as any irregularity in the wheel or want of alignment in the axle will cause differences in the readings for a given circle, or an alteration in the constant for various sizes of circles and different positions of the fixed point

Figs. 73 to 87 give the indicator cards obtained in the several tests. Tables V. to XVIII. give the results in detail obtained in measuring the indicator cards, the scales employed being those corresponding to about one-half the initial pressure.

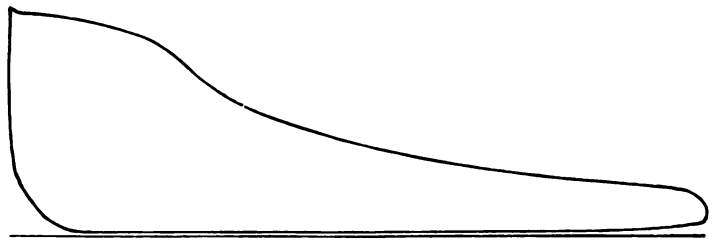


FIG. 73.

New Form of Indicator A.
Test No. 1. Table V.
Card No. 6. Rev. = 192. Steam pressure = 103 lbs.
Hot scale of spring = 81.7 lbs.

FIG. 74.

Old Form of Indicator A in ordinary condition.
Test No. 2. Table VI.
Card No. 7. Rev. = 197. Steam pressure = 96 lbs.
Hot scale of spring = 58.8 lbs.

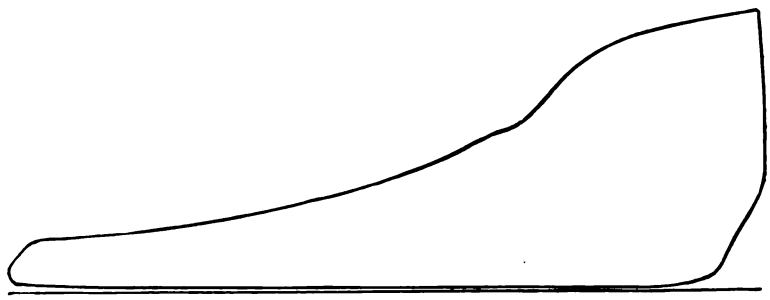


FIG. 75.

Indicator B.
Test No. 2. Table VI.
Card No. 7. Rev. = 197. Steam pressure = 96 lbs.
Hot scale of spring = 58.6 lbs.

FIG. 76.

Indicator C.
Test No. 3. Table VII.
Card No. 1. Rev. = 199. Steam pressure = 95 lbs.
Hot scale of spring = 48.8 lbs.

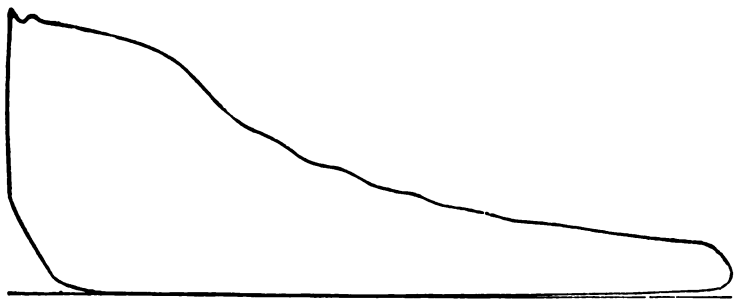


FIG. 77.

Old Form of Indicator A with weight attached to piston.
Test No. 5. Table IX.
Card No. 3. Rev. = 189. Steam pressure = 94 lbs.
Hot scale of spring = 58.8 lbs.

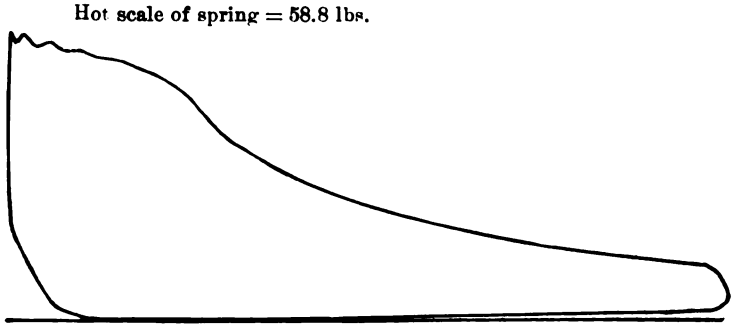


FIG. 78.

Old Form of Indicator A with loose piston.
Test No. 7. Table XI.
Card No. 4. Rev. = 196. Steam pressure = 96 lbs.
Hot scale of spring = 59.9 lbs.

FIG. 79.

Old Form of Indicator A with tight piston.
Test No. 8. Table XII.
Card No. 1. Rev. = 195. Steam pressure 95 lbs.
Hot scale of spring = 58.8 lbs.

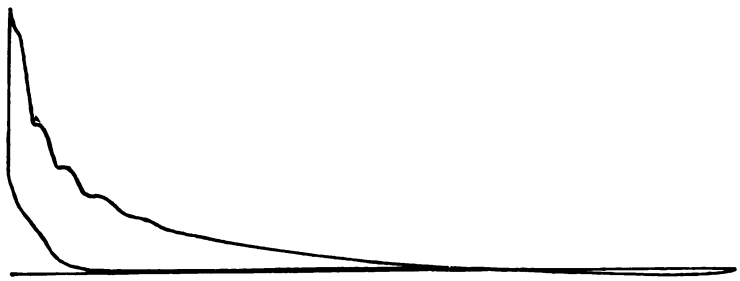


FIG. 80.

Old Form of Indicator A in ordinary condition.
Test No. 9. Table XIII.
Card No. 1. Rev. = 184. Steam pressure = 93 lbs.
Hot scale of spring = 58.8 lbs.

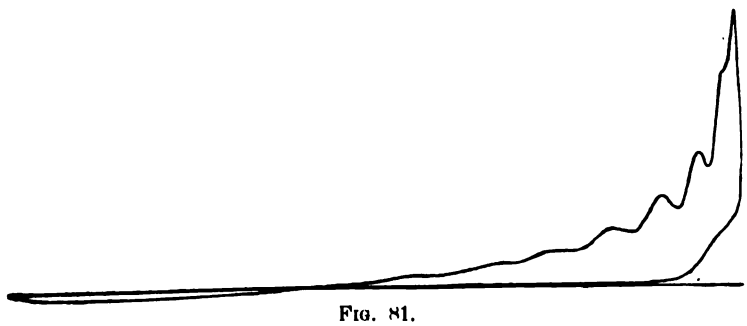


FIG. 81.

Indicator B.
Test No. 9. Table XIII.
Card No. 1. Rev. = 184. Steam pressure = 93 lbs.
Hot scale of spring = 58.6 lbs.

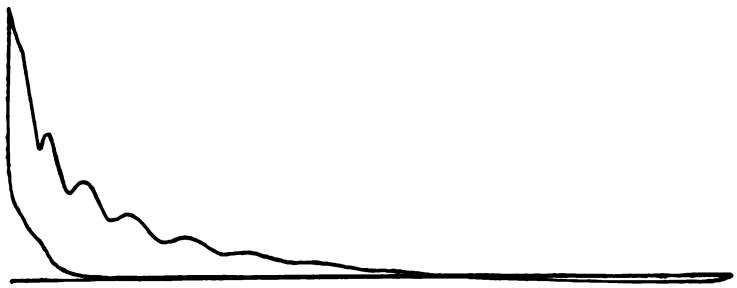


FIG. 82.

Old Form of Indicator A with weight attached to piston.
Test No. 10. Table XIV.
Card No. 6. Rev. = 178. Steam pressure = 96 lbs.
Hot scale of spring = 58.8 lbs.

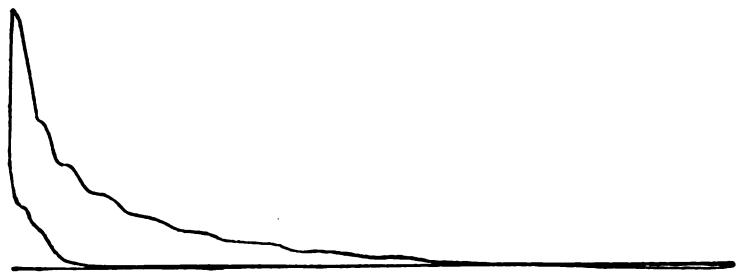


FIG. 83.

Old Form of Indicator A with tight piston.
Test No. 11. Table XV.
Card No. 2. Rev. = 176. Steam pressure = 93 lbs.
Hot scale of spring = 58.8 lbs.

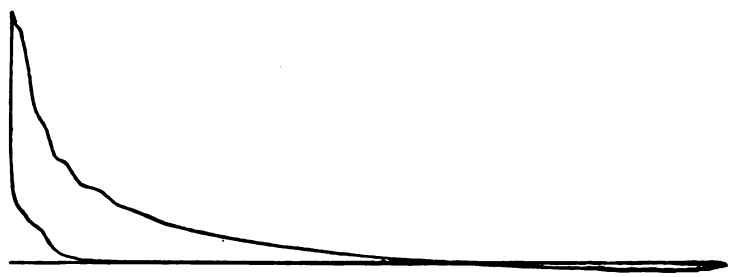


FIG. 84.

Old Form of Indicator A with loose piston.
Test No. 12. Table XVI.
Card No. 6. Rev. = 170. Steam pressure = 92 lbs.
Hot scale of spring = 59.9 lbs.

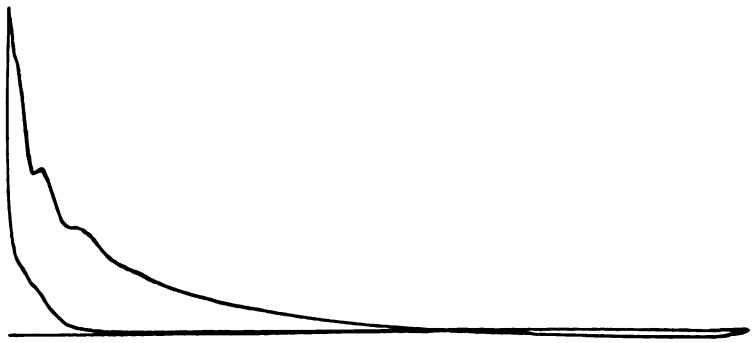


FIG. 85.

Indicator C.
Test No. 13. Table XVII.
Card No. 1. Rev. = 176. Steam pressure = 94 lbs.
Hot scale of spring = 48.8 lbs.

FIG. 86.

New Form of Indicator A.
Test No. 14. Table XVIII.
Card No. 6. Rev. = 176. Steam pressure = 94 lbs.
Hot scale of spring = 81.7 lbs.

FIG. 87.

Old Form of Indicator A in ordinary condition.
Test No. 14. Table XVIII.
Card No. 3. Rev. = 176. Steam pressure = 97 lbs.
Hot scale of spring = 58.8 lbs.

TABLE V.

Mean Effective Pressures of Cards Taken Simultaneously.

New form of A Indicator and Old Form of A Indicator. Pistons of indicators in ordinary condition. Test No. 1, Cut-off = $\frac{1}{4}$. October 16, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator New A. Scale = 81.7		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	104	192	.538	44.0	.742	43.6	+0.4
2.....	104	190	.550	44.9	.770	45.3	−0.4
3.....	104	189	.563	46.0	.766	45.0	+1.0
4.....	103	202	.571	46.6	.791	46.5	+0.1
5.....	108	190	.567	46.3	.799	47.0	−0.7
6.....	103	192	.554	45.3	.783	46.0	−0.7
7.....	108	185	.551	45.0	.771	45.3	−0.3
8.....	106	190	.567	46.3	.793	46.6	−0.3
9.....	101	191	.538	44.0	.765	45.0	−1.0
10.....	106	192	.563	46.0	.791	46.5	−0.5
11.....	106	193	.567	46.3	.792	46.6	−0.3
12.....	103	190	.547	44.7	.774	45.5	−0.8
Average ..	104.2	191.3	.556	45.4	.778	45.7	−0.3

TABLE VI.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Pistons in ordinary condition. Test No. 2, Cut-off = $\frac{1}{4}$. October 16, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	96	197	.696	40.8	.711	41.8	−1.0
2.....	95	188	.676	39.6	.678	39.9	−0.3
3.....	95	195	.702	41.1	.701	41.2	−0.1
4.....	92	184	.665	39.0	.656	38.6	+0.4
5.....	95	190	.707	41.4	.696	40.9	+0.5
6.....	95	190	.707	41.4	.706	41.5	−0.1
7.....	96	197	.680	39.8	.675	39.7	+0.1
8.....	96	197	.689	40.4	.688	40.5	−0.1
9.....	93	193	.673	39.4	.668	39.3	+0.1
10.....	90	181	.618	36.2	.619	36.4	−0.2
11.....	98	188	.640	37.5	.649	38.2	−0.7
12.....	98	190	.648	38.0	.648	38.2	−0.2
Average ..	94.9	190.8	.675	39.6	.675	39.7	−0.1

TABLE VII.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator C and Old Form of A Indicator. Pistons of indicators in ordinary condition. Test No. 3, Cut-off = ¼. October 16, 1893.

No. of Card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator C. Scale = 48.8.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1	95	199	.817	39.9	.670	39.4	+0.5
2	96	210	.794	38.7	.665	39.1	−0.4
3	93	209	.787	38.4	.649	38.2	+0.2
4	84	190	.746	36.4	.608	35.8	+0.6
5	87	190	.737	36.0	.621	36.5	−0.5
6	90	191	.765	37.3	.621	36.5	+0.8
7	86	186	.677	33.0	.559	32.9	+0.1
8	88	193	.688	33.6	.568	33.4	+0.2
9	92	192	.718	35.0	.597	35.1	−0.1
10	92	193	.698	34.1	.582	34.2	−0.1
11	92	192	.721	35.2	.600	35.3	−0.1
12	93	193	.702	34.3	.581	34.2	+0.1
Average	90.7	194.8	.738	36.0	.610	35.9	+0.1

TABLE VIII.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A indicator. Test No. 4, Cut-off ¼, October 20, 1893.

No. of card.	Steam pressure in lbs. per sq. in. above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1	95	193	.708	41.5	.703	41.3	+ 0.2
2	95	190	.704	41.3	.699	41.1	+ 0.2
3	95	190	.707	41.4	.710	41.7	− 0.3
4	94	188	.700	41.0	.695	40.9	+ 0.1
5	93	190	.685	40.1	.678	39.9	+ 0.2
6	98	193	.698	40.9	.700	41.2	− 0.3
Average	95.0	190.7	.700	41.0	.698	41.0	0

TABLE IX.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Indicator Old Form A. Weight attached to Piston of Old A Indicator. Test No. 5, Cut-off = ¼, October 20, 1893.

No. of card.	Steam pressure in lbs. per sq. in. above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1	96	188	.697	40.8	.691	40.6	+ 0.2
2	96	180	.719	42.1	.715	42.0	+ 0.1
3	94	189	.700	41.0	.697	41.0	0
4	94	199	.694	40.7	.691	40.6	+ 0.1
5	93	200	.686	40.2	.692	40.7	− 0.5
6	92	192	.676	39.6	.677	39.8	− 0.2
7	93	189	.686	40.2	.680	40.0	+ 0.2
8	94	198	.690	40.4	.688	40.5	− 0.1
9	93	192	.687	40.3	.686	40.3	0
10	95	191	.695	40.7	.703	41.3	− 0.6
11	98	189	.721	42.3	.730	42.9	− 0.6
12	93	198	.709	41.5	.711	41.8	− 0.3
Average	94.2	192.1	.697	40.8	.697	41.0	− 0.2

TABLE X.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Pistons of Indicators in Ordinary Condition. Test No. 6, Cut-off = ¼, October 24, 1893.

No. of card.	Steam pressure in lbs. per sq. in. above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1	94	195	.698	40.9	.702	41.3	− 0.4
2	93	196	.672	39.4	.677	39.8	− 0.4
3	94	195	.678	39.7	.672	39.5	+ 0.2
4	93	190	.676	39.6	.666	39.2	+ 0.4
5	96	204	.692	40.6	.691	40.6	0
6	96	198	.705	41.3	.698	41.0	+ 0.3
Average	94.3	196.3	.687	40.2	.684	40.2	0

TABLE XI.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Loose Piston in Old A Indicator Test No. 7, Cut-off = $\frac{1}{4}$, October 24, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 59.9.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	94	195	.696	40.8	.690	41.3	− 0.5
2.....	95	195	.699	41.0	.689	41.3	− 0.3
3.....	93	194	.692	40.6	.679	40.7	− 0.1
4.....	96	196	.717	42.0	.699	41.9	+ 0.1
5.	92	190	.681	39.9	.665	39.8	+ 0.1
6.....	93	190	.689	40.4	.677	40.6	− 0.2
7.....	90	186	.662	38.8	.655	39.2	− 0.4
8.....	100	197	.683	40.0	.678	40.6	− 0.6
9.....	98	200	719	42.1	.711	42.6	− 0.5
10.....	91	185	.665	39.0	.654	39.2	− 0.2
11.....	95	190	.681	39.9	.676	40.5	− 0.6
12.....	97	197	.703	41.2	.700	41.9	− 0.7
Average..	94.5	192.9	.691	40.5	.681	40.8	− 0.3

TABLE XII.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form Indicator A. Tight Piston in Old A Indicator. Test No. 8, Cut-off = $\frac{1}{4}$, October 24, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1... . .	95	195	.679	39.8	.675	39.7	+ 0.1
2.....	93	190	.677	39.7	.668	39.3	+ 0.4
3.....	92	194	.657	38.6	.650	38.2	+ 0.4
4.....	96	202	.693	40.6	.692	40.6	0
5.....	95	200	.680	39.8	.676	39.7	+ 0.1
6.....	92	193	671	39.3	.669	39.3	0
7.....	95	190	.683	40.0	.669	39.3	+ 0.7
8.....	95	190	.684	40.1	.684	40.2	− 0.1
9.....	96	190	.691	40.5	.681	40.0	+ 0.5
10.....	92	186	.662	38.8	.662	38.9	− 0.1
11.....	93	190	.666	39.0	.663	39.0	0
12.....	97	190	.690	40.4	.688	40.5	− 0 1
Average..	94.2	192.5	.678	39.7	.673	39 6	+ 0.1

TABLE XIII.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Pistons of indicators in ordinary condition. Test No. 9, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	93	184	.113	6.6	.123	7.2	−0.6
2.....	94	186	.117	6.9	.123	7.2	−0.3
3.....	93	180	.116	6.8	.126	7.4	−0.6
4.....	92	180	.110	6.4	.117	6.9	−0.5
5.....	95	178	.110	6.4	.123	7.2	−0.8
6.....	94	176	.112	6.6	.123	7.2	−0.6
Average ..	94.3	180.7	.113	6.6	.122	7.2	−0.6

TABLE XIV.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Weight attached to piston of Old A Indicator. Test No. 10, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	93	176	.099	5.8	.110	6.5	−0.7
2.....	96	177	.112	6.6	.127	7.5	−0.9
3.....	94	178	.112	6.6	.124	7.3	−0.7
4.....	94	175	.104	6.1	.109	6.4	−0.3
5.....	96	176	.115	6.7	.124	7.3	−0.6
6.....	96	178	.116	6.8	.125	7.4	−0.6
Average ..	94.8	176.7	.110	6.4	.120	7.1	−0.7

TABLE XV.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicators B and Old Form of A Indicator. Tight piston in Old A Indicator. Test No. 11, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	92	176	.115	6.7	.134	7.9	−1.2
2.....	93	176	.114	6.7	.138	8.1	−1.4
3.....	96	175	.120	7.0	.144	8.5	−1.5
4.....	92	174	.101	5.9	.118	6.9	−1.0
5.....	94	176	.109	6.4	.138	8.1	−1.7
6.....	96	176	.113	6.6	.146	8.6	−2.0
Average..	93.8	175.5	.112	6.6	.136	8.0	−1.4

TABLE XVI.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator B and Old Form of A Indicator. Loose piston in Old A Indicator Test No. 12, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator B. Scale = 58.6.		Indicator Old A. Scale = 59.9.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	92	172	.115	6.7	.117	7.0	−0.3
2.....	90	170	.108	6.3	.111	6.6	−0.3
3.....	88	169	.102	6.0	.103	6.2	−0.2
4.....	92	168	.114	6.7	.113	6.8	−0.1
5.....	90	170	.108	6.3	.113	6.8	−0.5
6.....	92	170	.116	6.8	.116	6.9	−0.1
Average..	90.7	169.8	.111	6.5	.112	6.7	−0.2

TABLE XVII.

Mean Effective Pressures of Cards Taken Simultaneously.

Indicator C and Old Form of A Indicator. Pistons of indicators in ordinary condition. Test No. 13, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator C. Scale = 48.8.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards.
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	94	176	.133	6.5	.120	7.1	−0.6
2.....	95	174	.136	6.6	.116	6.8	−0.2
3.....	97	178	.142	6.9	.125	7.4	−0.5
4.....	95	178	.159	7.8	.133	7.7	+0.1
5.....	93	174	.136	6.6	.118	6.9	−0.3
6.....	92	170	.144	7.0	.117	6.9	+0.1
Average..	94.3	174.2	.142	6.9	.122	7.1	−0.2

TABLE XVIII.

Mean Effective Pressures of Cards Taken Simultaneously

New Form of A Indicator and Old Form of A Indicator. Pistons of indicators in ordinary condition. Test No. 14, Cut-off = $\frac{1}{40}$. October 27, 1893.

No. of card.	Steam pressure in lbs. per sq. inch above atmosphere.	Revolutions per minute.	Indicator New A. Scale = 81.7.		Indicator Old A. Scale = 58.8.		Difference between M. E. P. of simultaneous cards
			Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	Mean height of card in inches.	Mean effective pressure in lbs. per sq. inch.	
1.....	98	176	.100	8.2	.129	7.6	+0.6
2.....	98	178	.095	7.8	.117	6.9	+0.9
3.....	97	176	.094	7.7	.124	7.3	+0.4
4.....	91	175	.082	6.7	.109	6.4	+0.3
5.....	91	175	.087	7.1	.111	6.5	+0.6
6.....	94	176	.086	7.0	.116	6.8	+0.2
Average..	94.8	176.0	.091	7.4	.117	6.9	+0.5

ADDED SINCE THE MEETING.

In the discussion of Professor Carpenter's paper, presented at this meeting, the theory has been advanced that the double line shown in many of the tests of indicators is due to what may be called a lag in the spring. That is, if the lines for a rising and falling pressure do not agree with each other, it has been

suggested that the fault may lie in the spring itself. In the tests of the springs used in my comparisons, no double lines were shown for a rising and falling pressure when tested with an accurately calibrated gauge.

In these tests the lag of the gauge itself was determined and allowed for. In the tests at the Brooklyn Navy Yard the lines for a rising and falling pressure agreed to within about one-sixty-fourth of an inch. The variation in this case was not accounted for definitely, but it may have been due to the action of inertia in the moving mercury column.

To determine the exact facts regarding the possibility of a lag in the spring, special tests have been made after the meeting, by applying a dead weight to the piston of an indicator and obtaining the lines for increasing and decreasing loads. To do this the indicator was held in a frame with the drum downward, and a rod was made to press against the piston, which was fitted with a special piece to hold the end of the rod in a central position. The rod was attached to a yoke at its upper end, which passed downward at each side of the frame. A pan for weights was attached to the yoke. The weight was thus made to hang, freely, directly from the centre of the piston, and compressed the spring while the latter was held in the same position it occupied during the tests with steam pressure. These tests showed that there was no double line produced with a rising and falling pressure, so that the lag due to the spring itself was less than could be appreciated in indicator measurements.

Tests were made with the spring cold, and also with the spring heated by steam.

It follows from these tests that indicators may be made to give the same results with rising and falling pressures. If, therefore, there is a difference found between the lines it should be traced out and eliminated, as this difference will produce an error in the power measurements.

APPENDIX.

METHOD OF CORRECTING FOR ALL VARIATIONS IN THE HOT SCALES OF THE SPRINGS.

As the scale of an indicator spring varies for different heights of the diagram, it is impossible to say which scale should be used by a simple inspection of the indicator card, and of the results of the tests to determine the hot scale of the spring. The general way we have adopted has been to employ the scale over the range of about the mean forward and the mean back pressure, but in the present investigation it was thought advisable to go further, and to employ the scales corresponding to each portion of the diagram.

The error of employing scales determined over any particular range has already been discussed. Thus, in the four indicators tested, if the scales had been determined over a range of 100 lbs. per square inch above the atmosphere, which was about 20 lbs. greater than the initial pressure, the error of so doing would have been as a maximum 3%, or, adding opposite signs, 5%. If the range had been from the atmosphere to the initial pressure, the error would have been as a maximum about 1%, or, adding opposite signs, 2%.

If the scales were determined between the mean forward and the mean back pressure, the error would have been only 1%, adding opposite signs, but this small variation is exceptional, and cases are often met in which the error is greater than this amount.

There is an element of uncertainty which enters in the use of an indicator spring, from the fact that it may not be at precisely the same temperature when employed in power measurements as it attains during the tests to determine the hot scale. This error would tend to affect all indicator springs in the same direction, so that it does not alter the comparative results shown by different indicators working under the same conditions, but it may be of importance in determining the absolute scales.

The total amount of the error is small, and will be made the subject of a future investigation.

The method of correcting for all variations in the scales is as follows:

Divide the card by a number of ordinates at equal distances apart, as represented in Fig. 88. Measure the areas A_1, etc., included between the forward pressure line and the atmospheric line, and the areas B_1, etc, between the back pressure line and the atmospheric line. Multiply each area by the scale corresponding to the average pressure, and let SA be the sum of the areas, multiplied by the scales, for the forward pressure line, and SB be the sum of the areas, multiplied by the scales, for the back pressure line; then the equivalent scale by which the

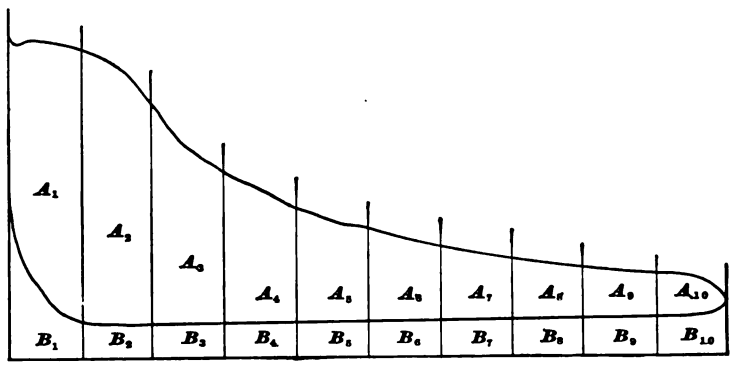


FIG. 88.

average height of the indicator card should be multiplied by in order to obtain the true mean effective pressure will be (SA – SB) divided by the area of the indicator card.

In calculating the equivalent scales, the largest and smallest cards of the set are taken, and if the scales for these two cards are nearly the same, the scales for intermediate cards may be determined by interpolation. If the scales are widely different for the largest and smallest cards, that of one or more intermediate cards should be calculated. In the present investigation, however, the equivalent scale of the largest and smallest cards of one set were found to agree within the limits of practical error, so that the equivalent scales for the other sets were taken as those corresponding to the average card. The results of the calculations are given in Tables XIX. to XXIV.

TABLE XIX.

INDICATOR OLD A.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

LARGEST CARD OF SET.

Division of card.	Area.	Mean Height.	Correspond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Correspond'g scale.	Area × scale.
A1	0.64	1.73	58.2	37.25	B1	0.06	0.16	56.8	3.41
A2	0.56	1.51	58.3	32.65	B2	0.02	0.05	56.8	1.14
A3	0.46	1.24	58.5	26.91	B3	0.01	0.03	56.8	0.57
A4	0.35	0.94	58.7	20.54	B4	0.01	0.03	56.8	0.57
A5	0.29	0.78	58.8	17.05	B5	0.01	0.03	56.8	0.57
A6	0.25	0.68	58.8	14.70	B6	0.01	0.03	56.8	0.57
A7	0.21	0.57	57.9	12.16	B7	0.01	0.03	56.8	0.57
A8	0.19	0.51	57.4	10.91	B8	0.01	0.03	56.8	0.57
A9	0.15	0.40	56.8	8.52	B9	0.02	0.05	56.8	1.14
A10	0.11	0.30	56.8	6.25	B10	0.03	0.08	56.8	1.70
Totals...	3.21	.....	.....	186.94	Totals...	0.19	.....	.....	10.81

Equivalent scale = (186.94 − 10.81) ÷ (3.21 − 0.19) = 58.3.

TABLE XX.

INDICATOR OLD A.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

SMALLEST CARD OF SET.

Division of card.	Area.	Mean Height.	Correspond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Correspond'g scale.	Area × scale.
A1	0.44	1.19	58.5	25.74	B1	0.07	0.19	56.8	3.98
A2	0.42	1.13	58.6	24.61	B2	0.01	0.03	56.8	0.57
A3	0.34	0.92	58.8	19.99	B3	0.01	0.03	56.8	0.57
A4	0.27	0.73	58.8	15.88	B4	0.01	0.03	56.8	0.57
A5	0.22	0.59	58.0	12.76	B5	0.01	0.03	56.8	0.57
A6	0.18	0.49	57.2	10.30	B6	0.01	0.03	56.8	0.57
A7	0.14	0.38	56.8	7.95	B7	0.01	0.03	56.8	0.57
A8	0.11	0.30	56.8	6.25	B8	0.02	0.05	56.8	1.14
A9	0.09	0.24	56.8	5.11	B9	0.02	0.05	56.8	1.14
A10	0.07	0.19	56.8	3.98	B10	0.02	0.05	56.8	1.14
Totals...	2.28	.....	.....	132.57	Totals...	0.19	.....	.....	10.82

Equivalent scale = (132.57 − 10.82) ÷ (2.28 − 0.19) = 58.3.

TABLE XXI.

INDICATOR OLD A, WITH LOOSE PISTON.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.
A1	0.54	1.56	59.0	31.86	B1	0.07	0.19	57.1	4.00
A2	0.48	1.30	59.5	28.56	B2	0.02	0.05	57.1	1.14
A3	0.43	1.16	59.7	25.67	B3	0.02	0.05	57.1	1.14
A4	0.31	0.84	59.9	18.57	B4	0.01	0.03	57.1	0.57
A5	0.25	0.68	58.8	14.70	B5	0.01	0.03	57.1	0.57
A6	0.19	0.51	57.6	10.94	B6	0.01	0.03	57.1	0.57
A7	0.17	0.46	57.2	9.72	B7	0.01	0.03	57.1	0.57
A8	0.15	0.40	57.1	8.56	B8	0.01	0.03	57.1	0.57
A9	0.13	0.35	57.1	7.42	B9	0.01	0.03	57.1	0.57
A10	0.09	0.24	57.1	5.14	B10	0.02	0.05	57.1	1.14
Totals...	2.74	.....	.....	161.14	Totals...	0.19	.....	.....	10.84

Equivalent scale = (161.14 − 10.84) ÷ (2.74 − 0.19) = 58.9.

TABLE XXII.

INDICATOR NEW A.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.
A1	0.42	1.13	80.7	33.89	B1	0.06	0.14	79.4	4.76
A2	0.40	1.08	80.9	32.36	B2	0.01	0.03	79.4	0.79
A3	0.35	0.94	81.4	28.49	B3	0.01	0.03	79.4	0.79
A4	0.24	0.65	81.7	19.61	B4	0.01	0.03	79.4	0.79
A5	0.21	0.57	81.0	17.01	B5	0.01	0.03	79.4	0.79
A6	0.17	0.46	80.1	13.62	B6	0.01	0.03	79.4	0.79
A7	0.14	0.38	79.8	11.17	B7	0.01	0.03	79.4	0.79
A8	0.12	0.32	79.4	9.53	B8	0.01	0.03	79.4	0.79
A9	0.10	0.27	79.4	7.94	B9	0.02	0.05	79.4	1.59
A10	0.04	0.11	79.4	3.18	B10	0.02	0.05	79.4	1.59
Totals...	2.19	.....	.....	176.80	Totals...	0.17	.....	.....	13.47

Equivalent scale = (176.80 − 13.47) ÷ (2.19 − 0.17) = 80.9.

TABLE XXIII.

INDICATOR B.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.
A1	0.56	1.46	58.5	32.76	B1	0.10	0.26	56.7	5.67
A2	0.54	1.40	58.6	31.64	B2	0.01	0.03	56.7	0.57
A3	0.45	1.17	58.6	26.37	B3	0.01	0.03	56.7	0.57
A4	0.33	0.86	58.6	19.34	B4	0.01	0.03	56.7	0.57
A5	0.25	0.65	57.9	14.48	B5	0.01	0.03	56.7	0.57
A6	0.21	0.55	57.1	11.99	B6	0.01	0.03	56.7	0.57
A7	0.17	0.44	56.7	9.64	B7	0.01	0.03	56.7	0.57
A8	0.15	0.39	56.7	8.50	B8	0.01	0.03	56.7	0.57
A9	0.13	0.34	56.7	7.37	B9	0.01	0.03	56.7	0.57
A10	0.08	0.21	56.7	4.54	B10	0.02	0.05	56.7	1.13
Totals...	2.87	.....	.....	166.63	Totals...	0.20	.....	.....	11.36

Equivalent scale = (166.63 − 11.36) ÷ (2.87 − 0.20) = 58.2.

TABLE XXIV.

INDICATOR C.

CALCULATION OF THE EQUIVALENT SCALE, CORRECTED FOR ALL VARIATION IN THE HOT SCALE OF THE SPRING.

Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.
A1	0.55	1.43	48.9	26.90	B1	0.11	0.29	47.9	5.27
A2	0.54	1.40	48.9	26.41	B2	0.02	0.05	47.9	0.96
A3	0.51	1.33	48.9	24.94	B3	0.01	0.03	47.9	0.48
A4	0.37	0.96	48.5	17.94	B4	0.01	0.03	47.9	0.48
A5	0.29	0.75	48.0	13.92	B5	0.01	0.03	47.9	0.48
A6	0.25	0.65	48.0	12.00	B6	0.01	0.03	47.9	0.48
A7	0.20	0.52	47.9	9.58	B7	0.01	0.03	47.9	0.48
A8	0.16	0.42	47.9	7.66	B8	0.01	0.03	47.9	0.48
A9	0.13	0.34	47.9	6.23	B9	0.01	0.03	47.9	0.48
A10	0.08	0.21	47.9	3.83	B10	0.02	0.05	47.9	0.96
Totals...	3.08	.....	.....	149.41	Totals...	0.22	.....	.....	10.55

Equivalent scale = (149.41 − 10.55) ÷ (3.08 − 0.22) = 48.6.

DISCUSSION.

Prof. R. C. Carpenter.—I wish to ask if these results given here are those which were obtained by test and not the nominal scale on the springs.

Professor Jacobus.—The results which I have given are calculated by employing the hot scale of the springs as determined

by tests at known steam pressures. If the nominal scales, as furnished by the makers, had been used, there would have been wide variations. For example, in one spring, for which the nominal scale is 50, the true scale, when all corrections are made, is 48.6 lbs., or there is a difference of about three per cent. With another spring the difference is one per cent. in the reverse way, so that the discrepancies which might arise by employing the nominal scales would be about four per cent.

In other springs which we have tested, aside from the present investigation, still greater differences have been found between the nominal and true scales. In all tests, therefore, the hot scales of the indicators should be determined and applied, and if this is done in the way indicated in the paper there should not be a discrepancy between the results of any two indicators of over two per cent.

Mr. Geo. I. Rockwood.—I had an interesting experience which showed the accuracy of construction of the Tabor indicator. While testing a triple engine I thought that the diagrams from the intermediate cylinder would look better if taken with 20-lb. instead of the 30-lb. springs such as I was using. I telegraphed to the makers to send me two 20-lb. springs, and on their arrival I tested them in the indicators. Although they were not selected with special reference to the particular pair of instruments I had, yet I could distinguish almost no appreciable error.

DLXXI.*

CRUCIBLE FURNACE FOR BURNING PETROLEUM.

BY W. E. CRANE, WATERBURY, CONN.
(Member of the Society.)

THE furnace is built round as high as the top of the crucible, and four or five inches larger in diameter. From there up it is built square, and the corners used as described further on.

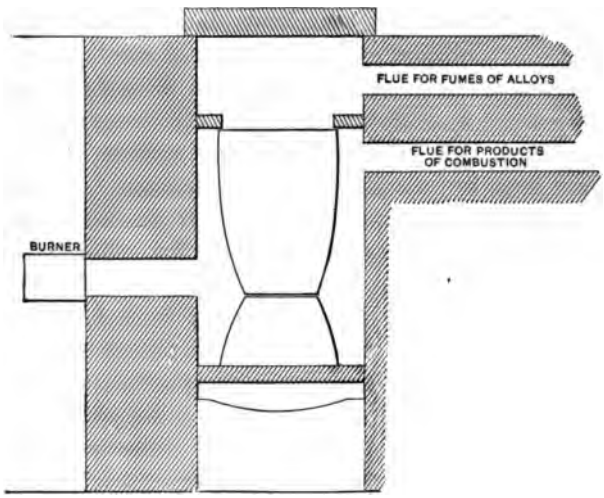


FIG. 89.

The pot stands on a support about six inches high. If it is a furnace running continually, it is immaterial of what substance it is made, or how solid, provided it will withstand the high temperature.

If a furnace runs ten to twelve hours only per day, it should be made with as little substance as will withstand the temperature and weight. If made solid, the first heat will take longer, as this support must be thoroughly heated through before the bottom of the pot will heat. After the first heat any support will be all

* Presented at the New York meeting (December, 1893) of the American So- of Mechanical Engineers, and forming part of Volume XV. of the *Transact*

right. Fig. 89 shows an elevation of the furnace, which has been rebuilt from one using grates for coal. A brick bottom is put over the grates to preserve them, but one grate should be taken out and a hole left through the bottom, so that the metal can run through if a pot breaks.

Fig. 90 shows a plan of furnace at the top of the pot. Four pieces should be provided of the shape shown in one of these corners, which will rest on top of the pot and in the corner of the furnace, shutting in the products of combustion around the pot below the top. By this means fully 15% of the fuel will be saved, a much hotter fire secured, and faster work done. A small flue should lead off from the top of this chamber, not over 2½ inches square, either to the chimney, or to a heater for heating the air, or to other system of economizing the heat from waste gases.

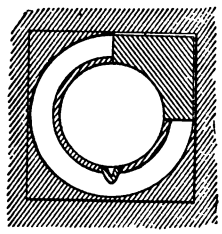


FIG. 90.

When the waste heat is utilized, this construction is essential in melting alloys, especially where spelter or zinc is used, as the fumes would coat over or fill up any economizer. By closing the space at the top of the pot the fumes from the alloys can be carried off through a separate flue. This flue should have from 30 inches to 40 inches area.

The burner would preferably be an air burner, with as low a pressure as possible. Some burners work nicely with as low as 6-ounce pressure. All of the air for combustion should be supplied by the blower. With an air burner at high temperatures there is usually a deposit of carbon. This can be prevented by introducing a small jet of steam, preferably superheated. As a general thing this will be found to cause more annoyance than that of removing the deposited carbon. The steam will have a tendency to lengthen the flame, and as a consequence the heat will be less intense, more of the heat passing off into the flue.

There should be a good draft to take off fumes when using alloys, which smoke a good deal; but the draft for the products of combustion should be light.

For the support for the crucible in furnaces running ten hours per day, the writer has found an old pot cut off at the right length and turned bottom up to be as good as anything tried.

The pieces to shut off at the top of the pot should be made of crucible stock and well seasoned.

DISCUSSION.

Mr. W. F. Durfee.—This paper has a good deal of interest for me. I merely wish to ask a couple of questions; one as to whether the sectoral piece of fire-clay on the top of your crucible interfered with pulling out the crucible from the melting hole; and the other, what the actual expense of melting with oil as fuel is, as compared with coal.

Mr. G. C. Henning.—I had occasion to watch the operation of some oil-fired crucibles for melting steel, and I found that the expense was neither in keeping the crucibles in repair nor in the waste of material, but it was altogether in the difficulty of keeping the floor of the shop or foundry in position, because it was always open to have the pipes and valves looked after and cleaned out, by reason of the deposit from the oil back of the furnace, on account of the heating of the floor and furnace walls around the pipes. What had to be done to overcome this constant necessity of repairing and cleaning the pipes and valves was to run a hot pipe along the oil-pipes to prevent any chilling after heating, and to provide a drip-system to gather all the heavy material out of the oil before it got anywhere near the furnace. Then, if, as in this process, pure oil is not used—it is probably intended to use crude Lima oil—there is so much deposit that even if the furnace stands for a short time, enough gases will be driven off to leave a heavy residue in valves and pipes, and if there is not a cistern, or collector, some distance from the furnace the pipes and valves will always become clogged, and the furnace will in a short time be useless, and it will cost a great deal to put it into good working order again.

Mr. A. A. Cary.—I would like to question the author as to the character of the flame which he uses in this furnace, and specially whether he admits air enough to produce a blue flame or whether he uses the radiant yellow flame. In looking into this question with a view to selecting burners for an oil-firing plant at the Columbian Exposition, I visited a number of oil furnaces and looked considerably into this matter. The blue flame is generally used in the laboratory or elsewhere where it is to come into direct contact with the article to be heated. But when it does not come into direct contact, and the yellow flame is used for radiation, it is apt to deposit carbon, as the author has stated.

The experience of the fuel oil department of the Standard Oil Company indicates that for the best results the flame should not strike the article directly, but against a baffle wall of fire-brick or something of the sort.

By the use of the yellow flame the effect of radiation is secured from the little red and white hot particles of carbon which form the flame, and by this means they get better results. By using the flame in direct contact with the article to be heated, they obtain only comparatively poor results.

Dr. H. G. Torrey.—I would like to ask the author whether he has experienced any bad effects on the crucible from its having a direct flame at one side only with no corresponding flame on the other? He also says the oil furnace should preferably be an air furnace. Is the other alternative the use of steam for bringing the air in, or can an oil furnace be run without a pressure of air at all?

What other burners are alluded to in saying "preferably an air burner"?

Mr. O. C. Woolson.—The construction of this furnace would indicate that it has a rather limited combustion chamber. It may be all right for the crucible furnaces; of these I have had but limited experience. But I have had considerable experience in larger heating furnaces, where it seemed to be very necessary to have a good, liberal-sized combustion chamber before the flame was brought in contact with the steel or with the metal in any way. It seems to me I should question a maximum efficiency of this furnace without some provision for a liberal combustion chamber contiguous to the crucible chamber.

*Mr. W. E. Crane.**—I will reply also to two criticisms which were made by a gentleman who is not able to be here to-day, one as to the economy and the other as to the life of the crucible. In this case the economy was about 30% over coal, at the price of $5 per ton paid for egg coal and two and a half cents per gallon paid for oil, at Waterbury, Conn. As to the life of the crucible, a crucible in a brass furnace will last about 35 heats, and in a silver furnace about 15. This is the crucible which we tried. We tried it on brass first and carried about 18 heats with brass, then we put the same crucible on to silver and cast about 14 heats from that. So that the life of the crucible

* Author's closure, under the Rules.

was perhaps about half used up ordinarily in heating brass, and then it had nearly the life of the crucible for melting silver. While we were conducting the experiments we got in a new lot of crucibles, and after trying those we found that on putting a new crucible in the furnace it would break; that the only way to get the crucible to stand was to get the first heat with coal and afterwards with oil. It was apparent that there must be a special mixture.

The pieces at the top of the pot, of course, would have to be removed. I know of no other way of shutting in anything of that kind. These pieces were put in in four sections, for the ease in removing them, and because in four pieces of that kind they were longer-lived than by having them in two pieces.

Mr. Durfee.—Did you have any trouble with pieces welding at the top of the pot?

Mr. Crane.—No. I never had any trouble with the welding at the top of the pot. The only trouble was at the bottom. This troubled only with the first heat, and was overcome by putting on some black-lead on the bottom of the pot before putting it in the furnace.

We experienced the same trouble that Mr. Henning mentions, of sediment collecting in the pipes, and also put in a settling-chamber to catch the deposit which comes from crude petroleum, and also the water. This not being sufficient, we arranged the pump so that the oil could be pumped out of the pipe leading to the furnaces; but with this the sediment could not be removed, so a steam-pipe was connected to the oil-pipe near the furnace, the oil pumped out, a valve turned so that it opened out-of-doors or into a catch-basin, and the pipe could be readily cleaned. A few trials of stopping-up would tell how often it occurred. It could then be anticipated, and the pipe blown out before it gave trouble. If a pipe was over 100 feet or 150 feet long, I should think it a good idea to have valves to divide it in sections, so as to avoid heating up too long a section at a time, and cracking fittings from too great expansion.

These furnaces were run with a white flame; that is the nearest we could get to it; possibly it might be called a yellow flame, but in this case it was white.

As to the size of the furnace, I think that in a furnace of this kind you could build it larger without any disadvantage. But as this was all the room that we had, the plan pursued worked very

nicely. I would say that in melting with coal at the draught they had for melting a 200 pot of 22% German silver and with a chimney that would give about .76 of an inch pressure of a column of water, it would take about two and a quarter hours after the first heat with coal. With oil, we have run them as fast as one in an hour. But running them faster your economy will decrease. If you run at a rate one-third faster than the coal, you will get no more economy than you would with coal. We had only this one furnace, and put this burner only on one side, and the pot is always set with this little nozzle or mouth towards the front, as that is the handiest way of getting the tongs on and carrying it to the molds, and we noticed no more deterioration in front of the crucible than in the rear. I have thought, however, if it was possible, that it would be better to put the burner in at an angle, so that it would go around instead of striking directly on. I say, that where it is possible to do that it would be preferable. I spoke of the air burner from having had experience on other work with the steam burner. The steam burner makes a longer flame, and for casting you want a very short flame and local heat, and have it intense in a small spot, which you probably could not get with a steam burner. We tried one burner of what used to be the Springfield or the Aerated system, which furnishes air rather from the current which is induced by the small steam jet which they have, and that was not a success. We could not make it work with that, and unless we had a very strong draught through the furnace, which would take in sufficient air for combustion, we could not regulate it. But with an air furnace, furnished from an ordinary fan, in which the full amount of air can be regulated, you could get any heat to run the furnace as fast or as slow as you chose.

DLXXII.*

ON THE MAXIMUM CONTEMPORARY ECONOMY OF THE HIGH-PRESSURE MULTIPLE-EXPANSION STEAM-ENGINE;

WITH A COMPARISON OF ITS EFFICIENCY WITH THAT OF ITS IDEAL REPRESENTATIVE UNDER SIMILAR EXTERNAL CONDITIONS.

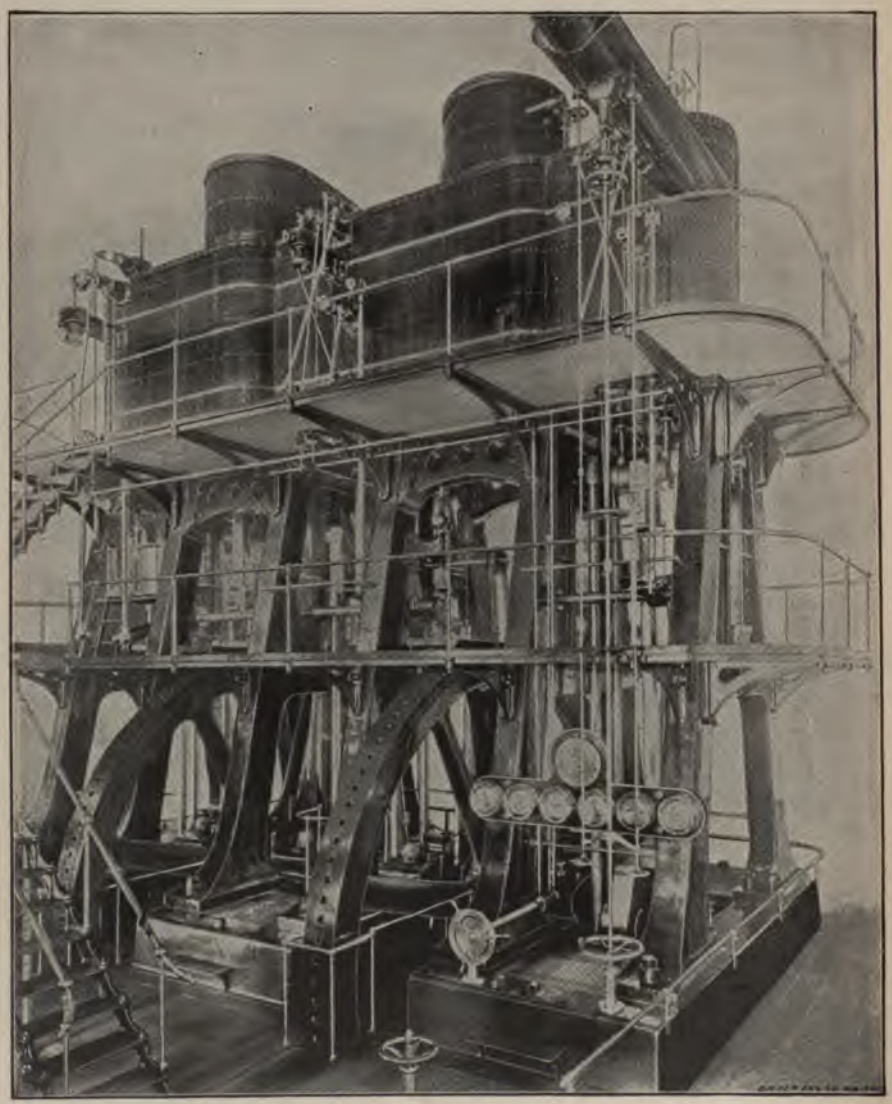
BY ROBERT H. THURSTON, ITHACA, N. Y.

(Member of the Society and Past President.)

THE MAXIMUM CONTEMPORARY ECONOMY OF THE STEAM-ENGINE of the time has now so nearly approached the computed efficiency of its ideal thermo-dynamic representative, under similar external working conditions, that it has come to be admitted that the distance separating them, a consequence of the more or less unavoidable and irreducible wastes to which the former is subject and from which the latter is free, although still growing less, is exceedingly difficult of further reduction. The closer the approximation of the real to the ideal, the more difficult is it further to improve the machine, and the more certain is it that any considerable and rapid gain is out of the question. We have, for a century, been constantly changing the conditions exterior to the machine, in such manner as to raise the maximum efficiency of the ideal case as steadily higher and more difficult of attainment by the real engine; while we have as constantly and steadily reduced the defects of the latter in such manner as to make the approximation of the real to the ideal all the time closer, in spite of these obstructing circumstances. Perhaps the best view of these changing conditions and performances may be obtained by observation of the continual gain exhibited in its history, since the days of Watt or of Newcomen, in the magnitude of the technically so-called "duty" of steam pumping engines.

With improving thermo-dynamic conditions and decreasing wastes in the engine, starting from the 5,000,000 duty of the

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions.*



THE MILWAUKEE TRIPLE-EXPANSION PUMPING ENGINE, 1892.

FIG. 91.

Savery engines still existing in the time of Watt, the 12,000,000 of Smeaton's Newcomen engine of the same period, and the first figures of Watt, perhaps averaging 20,000,000 foot-pounds per 100 lbs. fuel; and tracing the improvement through the most flourishing period of Watt's work, when he attained about 30,000,000, and his final perfection of the later Cornish engines, which, still later, attained in ordinary operation 60,000,000 to 80,000,000, we come to the period of successful introduction of the modern forms of the high-pressure multiple-expansion pumping engine, from 1860, giving duties ranging up to about 100,000,000, to 110,000,000, and to 120,000,000 in the succeeding generation and to date.

The improving thermo-dynamic conditions which have made these results possible have been, first, the gradual elevation of the steam pressure from 5 lbs. per square inch in 1800, to 20 lbs. in 1840, to 50 at the middle of the century, 75 a quarter of a century later, to 120 and to 150 and 175 in contemporary stationary and marine practice, and even to above 15 atmospheres in some instances. The second, and no less essential element of this progress, has been the simultaneous rise in the ratio of expansion from the time of Watt to the present, and from unity in his earliest practice to four for the latest Cornish form of the Watt engine, to six and eight a generation ago, and to fifteen and even twenty and more in the latest multiple-expansion machines. A terminal absolute pressure of about one-third of an atmosphere probably represents the limit to which expansion has been successfully carried. A higher rather than a lower terminal pressure is usual in the best practice of the day. The third element of improvement has been the increase of speed of piston and of rotation; although this has been less observable in steam pumping engines than in other types. Beginning with 200 to 300 feet per minute speed of piston, the figure has gradually risen to 500 and 600 in later years, and to above 1,000 in many cases to-day. These three have been the essential elements of improvement in the real engine; the latter supplementing the two others, which give thermo-dynamic gain. Improvements in design and construction have completed the advances thus made possible.

THE CULMINATION OF THIS CENTURY OF IMPROVEMENT, so far as the writer is at present informed, has been reached in the case to be here reported as illustrating the maximum contemporary economy of the high-pressure multiple-expansion steam pumping engine;

which case is to be compared with the representative thermo-dynamic case; the purpose being (1) to indicate what is the limit attained in this advance, up to date; (2) to show how far this limit is still removed from the corresponding limit of the pure thermo-dynamic case, and the location of the latter limit; and (3) the nature and extent of the wastes in the real case, the sum of which constitutes the difference between the real and the ideal cases. This latest and highest record of efficiency and duty, one which may very possibly long stand as the highest record—the passage of the date at which the quantity of steam used per hour and per horse-power is below 12 lbs. (5.5 kilograms), constituting an important era in the history of the steam-engine—is, on the customary American basis of 100 lbs. of fuel, 143,306,470 foot-pounds. On the old British basis of 112 lbs. (1 cwt.) it becomes 152,630,000, and on the proposed bases of 1,000 lbs. feed-water converted into dry steam, and of 1,000,000 B. T. U., is 154,048,000 and 137,656,000, respectively. For kilogrammeters of work per kilogram of fuel, the figure becomes 429,110. These figures, compared with those of the preceding paragraph, will complete the record of progress to date, and perhaps indicate practically the limit of advance for the nineteenth century in this department of human achievement. It still leaves us a margin of nearly 25% between the ideal and the real case for further gain; the thermo-dynamic case, assuming similar pressures and ratio of expansion, demanding about 9 lbs. of steam per horse-power per hour, and giving a duty of close upon 175,000,000 on the first-named basis, and about 200,000,000 on the highest standard. The ideal and the real efficiencies correspond, respectively, to fuel consumptions of 1.1 and 1.8 lbs. per horse-power per hour, with good boilers.

The Marquis of Worcester, more than two centuries ago, announced his modernization of the Hero steam-fountain as affording "an admirable and most forcible way to driue vp water with fyre," and "with little charge to drayne all sorts of mynes" (1659), but he little thought that he was utilizing perhaps only one per cent. of the thermo-dynamically transformable thermal energy of his fuel, or that the steam-engine, at the end of the nineteenth century, would "driue vp water with fyre" with "little charge to drayne all sorts of mynes," at the rate of a penny where it cost him three shillings. No one, in his time, dreamed that his marvelous "fire-engine" wasted 100 lbs. of fuel where it utilized one, or that the wastes of the engine were so largely capable of reduction by

the inventive and constructive genius of the mechanic and engineer.

OUR "REAL CASE," THE MILWAUKEE PUMPING ENGINE, built for that city by the E. P. Allis Co. from the designs of Mr. Reynolds, has been in operation since August, 1891, doing continuously so high a duty as to place it among the most remarkable constructions of its class and time, and probably to make its record the highest on record to date. It draws water from Lake Michigan and forces it into the city mains under a head of about 160 feet, into a reservoir, from which other engines raise it into a still higher level, so far as needed for consumption at greater elevations. The engine here under consideration is at the North Point station, at the level of the lake, very nearly. The station contains also a pair of compound engines, built in 1874, which gave a duty, originally, without cut-off on the low-pressure cylinder, of 77,000,000 on 100 lbs. of coal. After fitting this cylinder with cut-off gear, the duty rose to above 85,000,000. The pump raises about 900 gallons per minute, with a slip of 2%. There is also, besides these, a compound engine by Allis & Co., built in 1884, delivering 12,000,000 gallons per day against a head of 150 feet, and with a duty reported by Mr. Benzenberg, the City Engineer, from test, of 104,820,431 foot-pounds per 100 lbs. of best coal. The latest of the collection is that here to be examined—a triple-expansion engine built to raise 18,000,000 gallons per day of twenty-four hours, guaranteed by its builders to perform a duty of not less than 125,000,000 per 100 lbs. of anthracite coal.*

THE HISTORY OF THE WORK is briefly as follows: The attention of the writer was first called to this remarkable case by Mr. E. D. Leavitt, who, in December, 1892, reported its performance as 12.17 lbs. of water per horse-power per hour, and suggested obtaining from its designer and builders the facts relating to the plant, with a view to publication in the interests of science and of the profession. Acting at once upon this suggestion, the writer secured permission from the City Engineer's office, and from the builders and the designers of the engine, to make a special duty-trial of the machine, in the interests of all concerned and of the profession especially. The trial was to be made by the Sibley College staff, as a part of the year's work of the Department of Experimental Engineering, with every facility that either the pro-

* See Appendix A for Mr. McMillan's report.

prietors, the builders, or Cornell University and Sibley College could furnish. The results were to be worked up in the Sibley College laboratories, reported by the college to the designers, builders, and users of the engine, and published as soon as practicable by the writer, acting for the parties interested.

It was proposed that the trial should be made during the Easter vacation of the college, by a party to be selected, organized, and directed by Professor R. C. Carpenter, and to include such skilled and trained observers as could best be secured from his own department, re-enforced by observers sent in by the City Engineer and by the Allis Co. Among the "crew" sent out from Ithaca were several men engaged in graduate work, seeking material for their "Masters' theses," and some undergraduates who had enjoyed exceptional opportunities and shown great skill and efficiency in work of this kind. The special desire of the writer was to secure such data as would serve for a comparison of the theoretical ideal and the real engine, such as is here attempted, which comparison had never before been made in precisely this manner or with what it was hoped would prove such unexampled completeness.

This trial was finally made as proposed, and the results proved to be even more striking and satisfactory than had been claimed by the builders or reported to the writer. The data and observations are preserved on the files of the Sibley College laboratory.

THE DESIGN OF THE ENGINE (Fig. 91) embodies some special features regarded by the designer and builders as important, and as contributing effectively to the attainment of this exceptional economy. According to the statement given by Mr. Irving Reynolds, to whom the writer is indebted for many facts of interest and useful information, the designer was guided to a certain extent by these considerations:

"Starting with proposition that an outside packed plunger pump is the simplest, and under most conditions the best form of pump, the next step is to use the least number that will give a uniform flow of water; viz.: three, set at equal angles (120°). Having three plungers, they can as well be single as double acting, as the flow of water is practically constant and the single acting is much the simpler form of pump." (Fig. 92.)

To these principles, as appertaining to the pump, the writer is inclined to add, as affecting the steam-end: the clearance volumes, and their surfaces especially, should be of minimum magnitude; the steam passages should be of such large section, and

the valves so adjusted, as to cause no loss of pressure in the introduction nor any perceptible loss of work in the rejection of the working charge; the jackets should be applied to heads even more scrupulously than to the sides, where, in fact, experience shows

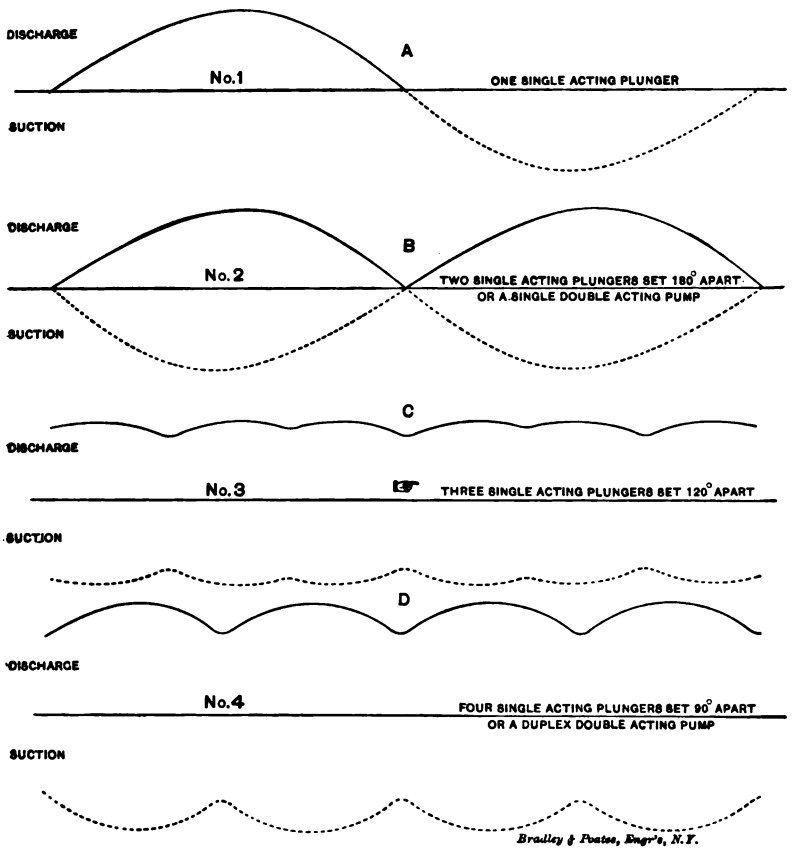


FIG. 92.—FLOW OF WATER IN PUMPS.

that they may be of little, perhaps in some cases of no, advantage.*

In the opinion of Mr. Reynolds, also: for multiple-expansion engines to do their best work requires that the conditions under which the engine is to work must remain constant for long periods, and that these conditions must be definitely known when design-

* The ideas of Mr. Edwin Reynolds in this matter are given in the writer's paper on Steam-jackets; *Trans. A. S. M. E.*, 1890.

ing the engine. The marine engine, on long voyages, illustrates these conditions, and a pumping engine on reservoir work has as nearly perfect conditions as any other machine, while the arrangements for obtaining high economy can be more readily supplied than at sea. Another reason for the use of triple expansion for pumping engines is, that having found three pumps desirable for the water ends, three steam cylinders become necessary, and the cost then is not increased by making them triple, rather than compound.

An interesting peculiarity of the practice of Mr. E. Reynolds has been the application of the principle of multiple expansion in larger degree for a stated pressure than is usual among other designers, as, for example, in his quadruple expansion engine with steam at 80 lbs. by gauge, at Warren, R. I. Mr. Irving Reynolds also furnishes the following interesting historical facts:

The first engines embodying these ideas were a pair of 5,000,000 gallon machines designed by Edwin Reynolds, and built for the water-works at Allegheny, Pa., in 1883. They were of the three-cylinder compound type, having one high-pressure and two low-pressure cylinders, connected to separate cranks set at angles of 120°. The steam-cylinders were set at the floor level, the crank-shaft and fly-wheels being carried on top of heavy A frames. The pumps were located beneath each steam-cylinder, the steam piston-rods being extended through the bottom cylinder-heads and connected to the pump plungers. These machines were very successful, developing a duty of 110,000,000 foot-pounds at 120 feet piston speed, with steam at 107 lbs. pressure.

Following the same line of construction this firm built, in 1886, what is believed to be the first triple-expansion engine applied to water-works service. The arrangement of the pumps was the same as with the compound engines described above, *i. e.*, three single-acting plungers * (Fig. 92), but the positions of the steam-cylinders and shaft were reversed, and cylinders were mounted on top of the A frames, the shaft being carried in journals in the bed-plates, similar to marine practice. The steam pressure specified by the city was not to exceed 80 lbs. per square inch, and predictions were freely made that with this low pressure the triple-expansion engine would be a failure as far as fuel economy

* The fluctuations of action are well shown in Fig. 92, for the single, the double, the triple, and the Worthington duplex systems of pumps. Case *C* is that here described. See, also, *Man. St. Eng.*, Vol. I., p. 177.

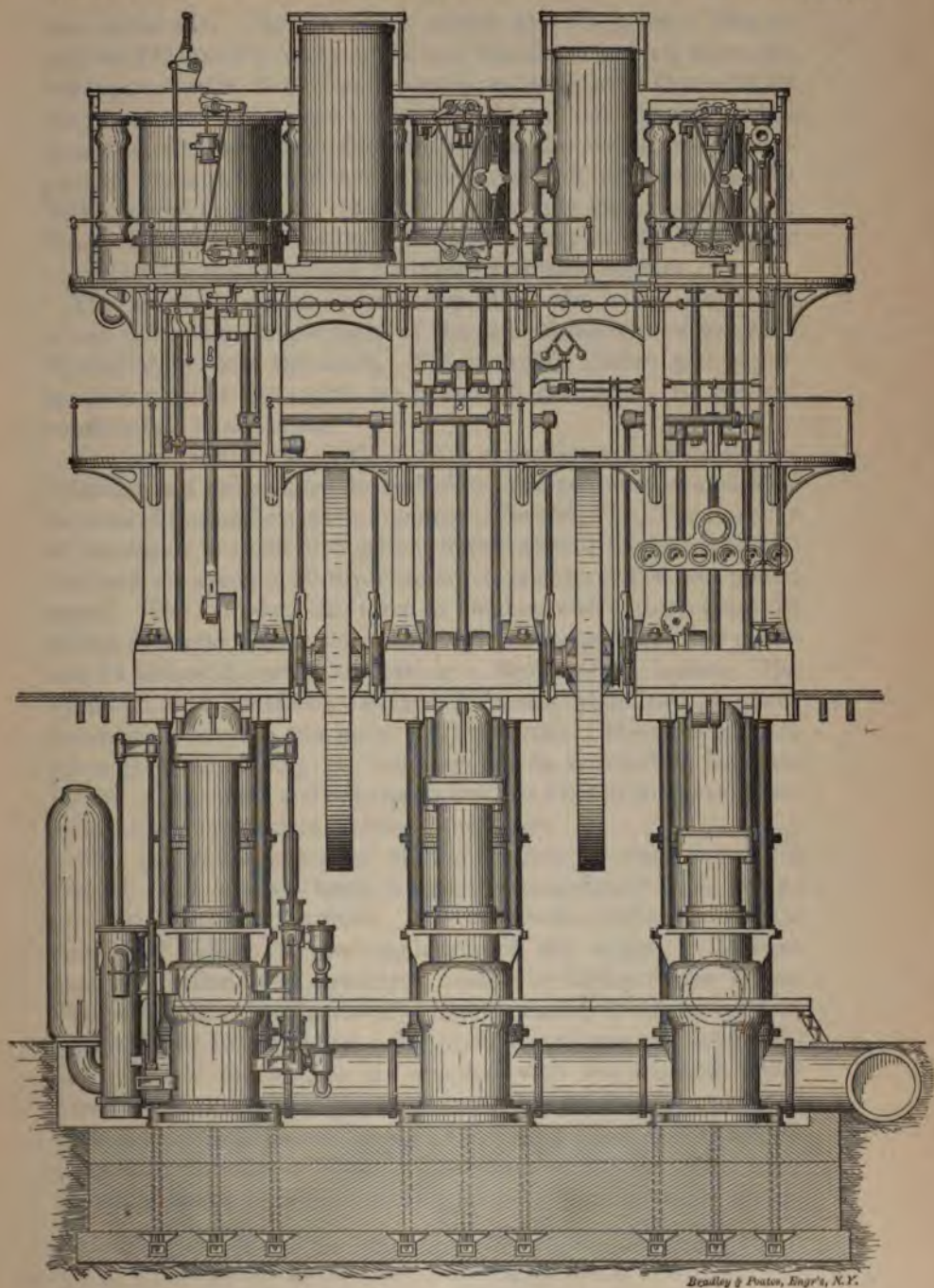


FIG. 93. FRONT ELEVATION OF ENGINE.

was concerned. This engine is stated to have given a duty as high as 129,000,000 foot-pounds, and has developed an indicated horse-power with 13.84 lbs. of steam per hour, with steam at 80 lbs. pressure; these results seeming to bear out Mr. Reynolds's idea that whatever gain there may be in the use of multiple-expansion engines, the percentage is as great with moderate as with higher steam pressures. In other words, while high pressures are desirable with almost any type of engine, they are not absolutely necessary for the success of multiple-expansion engines.

The Milwaukee engine was designed by Mr. Irving Reynolds, under the general supervision of the superintendent of the Allis Works, Mr. Edwin Reynolds. The pumping station and a perspective view of the engine are presented herewith. Its general construction is as follows:

The engine is of the vertical triple-expansion type, with steam-cylinders and parts above the bed-plate arranged somewhat similar to those of the modern marine engine. (Figs. 93, 94.) The capacity of the engine is 18,000,000 gallons in twenty-four hours, raised 160 feet, and the speed is 20 revolutions per minute, or 200 feet piston speed. The cylinders are three in number, one high-pressure, 28 inches diameter, one intermediate, 48 inches, and one low-pressure, 74 inches diameter, all having a stroke of 60 inches. The cylinders are mounted on cast-iron A frames, which rest on heavy bed-plates carrying the main shaft journals. The cylinders are steam-jacketed, having the working barrels inserted as separate pieces. The steam and exhaust valves are located in the cylinder-heads, their chests thus jacketing the latter.

The general appearance of the engine is well shown in Fig. 91, as it appears from the engine-room floor.* The pumps are entirely below the floor. The fly-wheels perform the office simply of aiding in the regulation of the engine. Between each two cylinders is a receiver, heated by high-pressure steam. The volumes are, respectively, high-pressure, 101.3 cubic feet, intermediate, 151 cubic feet. In the high-pressure and intermediate cylinders the jackets are supplied with live steam at boiler pressure; but on the low-pressure cylinder the jacket is supplied with steam which passes through a reducing valve, and which has a constant pressure of 34 lbs. The steam is supplied to the high-pressure jacket by a pipe leading directly from the main steam-

* Thurston's *Manual of the Steam-Engine*, Vol. I., p. 176.

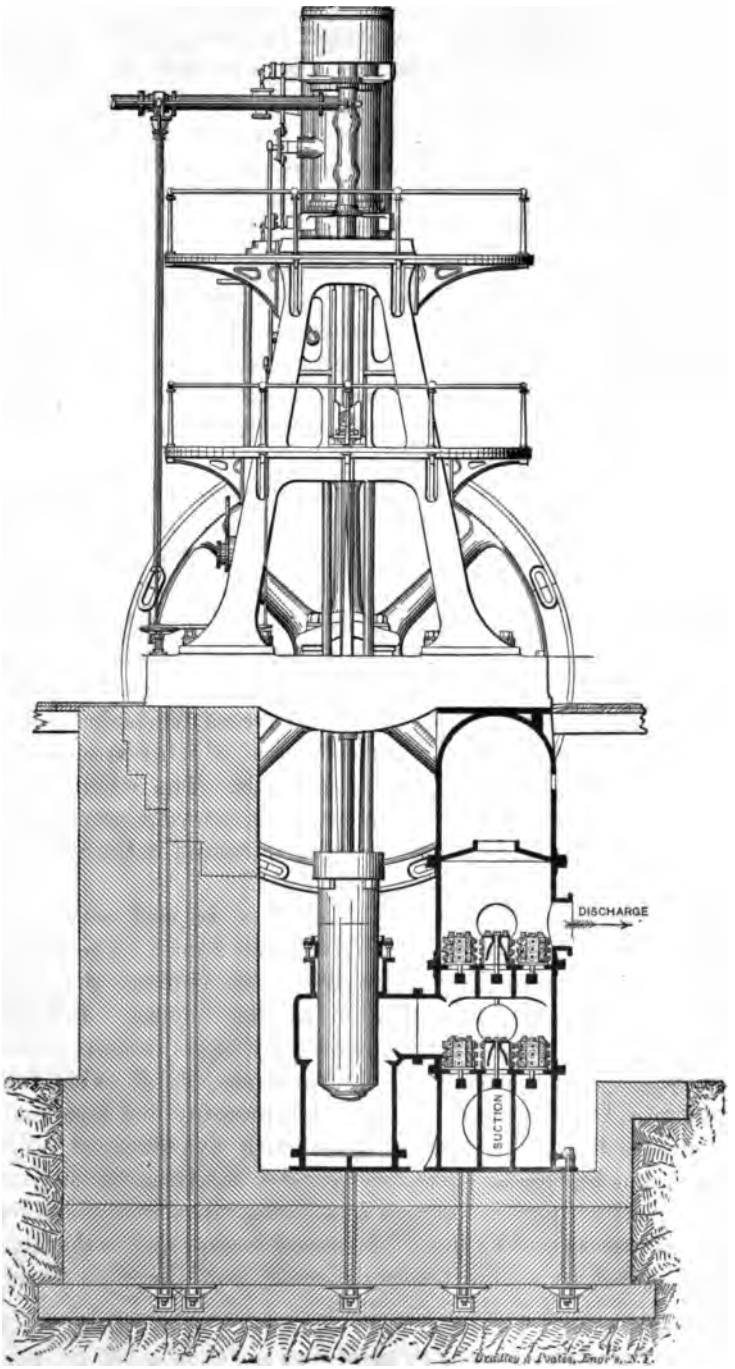


FIG. 94.--SECTION OF PUMPS.

pipe; the exhaust from this jacket supplies the jacket of the intermediate cylinder, and this is led into a trap, the overflow of which ordinarily passes into the suction of the feed-pump for supplying the boilers. The exhaust steam from the low-pressure cylinder jacket is similarly received into a steam-trap and discharged into a feed-pipe.

The exhaust valves of the low-pressure cylinders when closed, are flush on the inside of the cylinder-head, thus eliminating clearance due to exhaust ports. All the valve ports are located in the cylinder heads.

The valve gear is of the Corliss type on the high and intermediate cylinders, and a combination of Corliss steam valves and poppet exhaust valves on the low-pressure cylinder. The clearances in the cylinders and ports are: H. P., $1\frac{4}{10}\%$; I. P., $1\frac{5}{10}\%$; L. P., $\frac{77}{100}$ of 1%.

The cut-offs of the high-pressure cylinder are under the control of a speed governor combined with an adjustable hand cut-off. The cut-offs of the intermediate and low-pressure are controlled by a hand gear only, the point of cut-off in these cylinders being set usually in the inverse ratio of the cylinder volumes. Each steam-piston has two rods connecting to a forged steel cross-head; from this cross-head extend four tie-rods to the head of the pump plunger, thus transmitting the motion direct from the steam-pistons to the pump plungers, without passing through links or beams of any kind, making the machine direct-acting in the best sense of the term.

The upper ends of the connecting rods take hold of the cross-head in the space between the two piston-rods, the lower end connecting to its crank-pin. There are three cranks set at angles of 120°, and the motion of each pump plunger being controlled by its own crank, tends to secure uniform flow of water through the pumps.

The center crank (intermediate) is considerably larger than the others, and is fitted with drag boxes, to prevent the breaking of the shaft in case it gets out of alignment. There are two fly-wheels, 20 feet diameter, and weighing 50,000 lbs. each.

The accompanying diagram (Fig. 95) by Professor Peabody shows the rotative effect of the reciprocating parts.

The pumps deliver only on the down stroke, but the plungers are balanced to one-half the head, so that the work in the steam-cylinders is uniform on both the up and down strokes, and the only

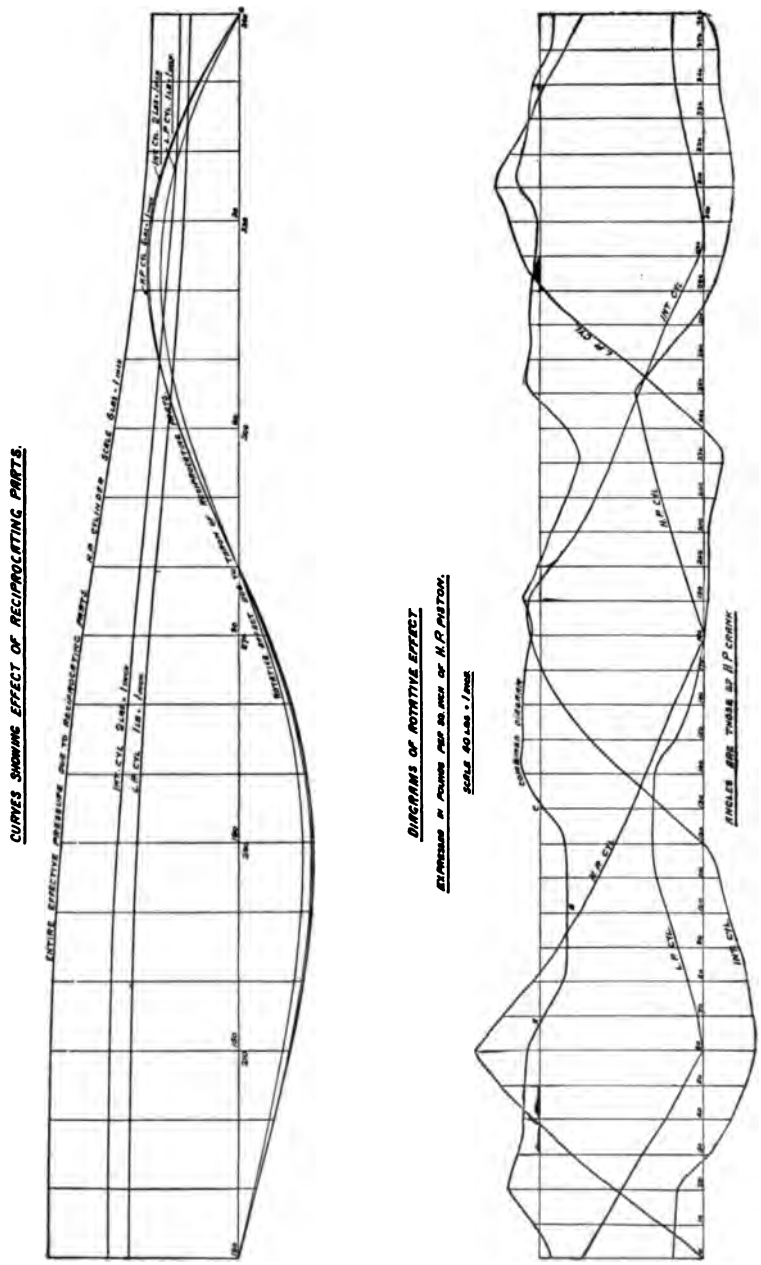

CURVES SHOWING EFFECT OF RECIPROCATING PARTS.
DIAGRAMS OF ROTATIVE EFFECT
ELEVENTH DIAGRAM
INT. CYL.
L.P. CYL.
H.P. CYL.

FIG. 95.

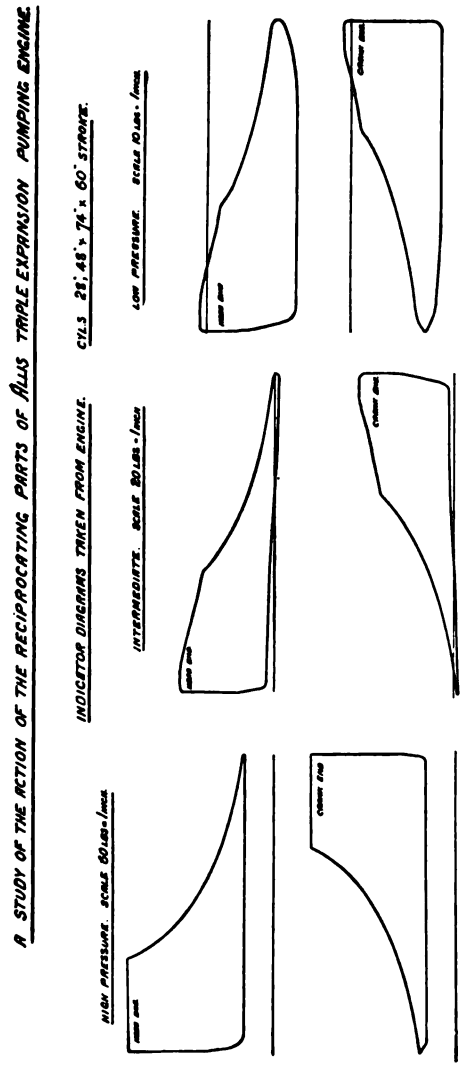


FIG. 96.

DIAGRAM OF ROTATIVE EFFECT.

Allis Triple Expansion Pumping Engine, North Point Pumping Station.

By Prof. C. H. Peabody, Boston, Mass.

load passing through the crank-shaft is that which is given to and taken from the fly-wheels at the beginning and end of each stroke.

The pumps are three in number, one located beneath each engine. The plungers (32-inch diameter) are of the single-acting outside-packed plunger type, each plunger being driven from a cross-head. The valve-chambers are cylindrical, and extend above the discharge valves, forming air-chambers, on which rest one end of the engine bed-plates, this latter arrangement saving foundations, and at the same time rendering the pumps easily accessible. The pump-valves are rubber, of small diameter ($3\frac{1}{2}$-inch), and are mounted in groups of 28, on "cages." There are seven of these cages (196 valves) in the suction and discharge of each pump, with an area about equal to that of the plunger.

The engine is fitted with a surface condenser, the air, feed, and circulating pumps being driven by an arm extended from the low-pressure plunger. The air-pump is 22 inches diameter by 60 inches stroke, and is single-acting.

STEAM IS SUPPLIED BY A BATTERY of five horizontal tubular externally fired boilers, with ordinary brick setting. Four boilers are ordinarily used, though three are ample for supplying this engine, four readily supplying steam for an additional 12,000,000 engine in the same building. The boilers are 66 inches diameter by 18 feet long, each containing 55 4-inch tubes, and having about 1,200 square feet of heating surface. The boilers have steam domes 32 inches diameter by 48 inches high, but have no mud-drums. The boilers are fed by a perforated pipe extending the whole length of the boiler above the tubes, and just below the surface of the water.

The arrangement of the pumps is shown in Fig. 97; a plan of engine foundation, pumps, boilers, and arrangements for supplying feed water. The pumps, *P*, are located directly beneath the engine cylinders. The valve-chambers are located in a cylinder, *VC*. Instead of being arranged on a single plane, they are situated on the sides and tops of hexagonal prisms. (Fig. 94.) The valves are rubber, about four inches in diameter, and their combined area is equal to that of the pump plunger. The water is received through the lower portion of the valve chamber, passing the suction valves, then is, by the reverse motion of the plunger, forced outward and upward past the delivery valves. The well is connected with Lake Michigan by a tunnel. The bottom is 39.8 feet from the pressure

gauge on the engine-room floor. The center of the valve-chamber is 19.8 feet below the gauge, and the suction lift from the center of valve-chamber to the surface of the water averaged 10.77 feet.

THE CONTRACTOR'S TRIALS of this triple-expansion engine were made in July, 1892, by Messrs. Benzenberg and Lewis; the first to determine the capacity of the engine, the second to secure a record of its maximum economy. The data and results of this series of trials, as supplied the writer, are as below. No determination of the quality of the steam was made; but it was undoubtedly substantially the same as in the Sibley College trial, to be described later. It is also very possible that the small leak discovered on the latter occasion then existed, and this may possibly account for a part, if not all, of the 3% difference in duty observed between the two cases. The longer working of the engine also reduced its friction somewhat, and this will account for some gain of duty, as, on the last trial, the machine exhibited an exceptionally excellent mechanical efficiency. That the latter condition has had a sensible influence is further indicated by the fact that on starting the engine at the beginning, with its full load, it was found that it could only work for a time, and until its bearings were smoothed up considerably, when prime steam was turned into the intermediate cylinder, to help the engine by thus increasing its power. The duty obtained during these tests, as a maximum, is, however, but about 3% less than that attained during the Sibley College trials, and its designers think it very possible to still further improve its performance somewhat by some slight changes and refinements of construction and operation. The following are the figures officially reported, and upon which the engine was accepted by the city of Milwaukee. Its regular work began August 23, 1891.

It was the outcome of this trial, as reported to the writer, which led to the later and more remarkable work which is the subject of this paper.

CONTRACTOR'S TRIALS, TRIPLE-EXPANSION PUMPING ENGINE.

	1892.	
Date of trial	July 12, 13.	July 20.
Duration, hours	24	24
Total revolutions of engine	29,566	29,524
Revolutions per minute	20.532	20.503
Average steam pressure by gauge	125.8	125.33
Average vacuum pressure by gauge	13.794	13.75

Average first receiver pressure by gauge	28.17	29.83
Average second receiver pressure by gauge	0.406	1.10
Barometer, inches	27.99	—
Total head of water pumped against	153.56	—
Average temperature of feed water, Fahr.	130°	132.2°
Total coal burned (Anthracite egg)	18,700	19,200
Total ashes	2,690	3,187
Per cent. of ashes and refuse	14.385	16.6
Actual evaporation per pound of coal	8.76	8.758
Equivalent evaporation from and at 212°	9.91	9.89
Coal per square foot of grate per hour	—	7.5
Total number of gals. pumped by plunger displacement	18,528,657	—
Average indicated horse-power	557.029	576.449
Total weight of feed-water	—	168,163
Feed water per I. H. P. per hour	—	12.155
Coal per I. H. P. per hour	—	1.387
Duty per 100 lbs. of coal burned, foot-pounds	126,865,240	—
Duty on supposition that evaporation was 10.27 from and at 212°, foot-pounds	132,950,524	—
Duty as above coal if containing 3% water	136,900,000	—

That this exceptional duty is not the result of accidental and fortunate concurrence of favorable conditions is proven by the fact that several other engines of similar design have closely approximated these figures. Thus, the trial of the three Chicago engines, by Messrs. R. W. Hunt & Co., has since given the following: *

RESULTS OF CONTRACTOR'S TRIAL OF CHICAGO PUMPING ENGINES, JULY, 1893.

Aggregate delivered horse-power	1,031.47
Mechanical efficiency	0.9143
Aggregate indicated horse-power	1,128.15
Feed water per delivered horse-power per hour	13.267
Feed water per indicated horse-power per hour, pounds	12.13*
B. T. U. per delivered horse-power per minute	244.6
B. T. U. per indicated horse-power per minute	223.7
Indicated horse-power per square foot of upper grate (H. furnace)	12.82
Heating surface per indicated horse-power	4.35
Foot-pounds per 1,000 lbs. of feed water (contract basis)	148,958,000
Foot-pounds per 1,000,000 B. T. U	134,643,030
Foot-pounds per 100 lbs. of bituminous coal	118,022,900
Foot-pounds per 100 lbs. of combustible	128,852,500

* No correction for moisture.

The comparatively low duty per hundred pounds of coal is due mainly to the grade of fuel, but is partly owing to the fact that only four boilers out of nine were in use, thus making the evaporation per square foot of heating surface too high for the best

* Report to City Engineer of Chicago, July 5, 1893.

results. These figures are reproduced here, not simply as of striking importance and interest, but also as showing the certainty that the still more remarkable figures here to be presented have been repeatedly approximated, and that there was, beforehand, no inherent improbability of their attainment.

The second of the older engines is usually operated in connection with the triple-expansion engine; their combined capacity (30,000,000) being about equal to the maximum daily consumption of water by the city.

THE ANNUAL DUTY OF THIS STATION, when all the work was done by compound engines, was about 80,000,000 foot-pounds per each 100 lbs. of coal burned, and for the year 1892, with the triple-expansion engines pumping five-eighths of the total water supply, the duty was nearly 100,000,000 foot-pounds per 100 lbs. of coal burned—a saving of 25%; and if the water could all be pumped by engines equal in efficiency to the triple-expansion engine, the average duty would be not less than 115,000,000 foot-pounds, a saving of more than 40% over the compound engines.*

The duty figures given include all coal burned during the year, for all purposes, including heating, steam for electric-light engine, and for running the Water-works repair-shop.

The following is Mr. McMillan's report of weekly records:

WEEKLY RECORD OF NO. 4 TRIPLE-EXPANSION PUMPING ENGINE, SHOP ENGINE RUNNING ON SAME BOILERS, MAKING REPAIRS AND LIGHTING BUILDINGS.

	Hours run.	Revolutions Made.	Coal Consumed. lbs.	Ashes. lbs.	Average Head of Water, in feet.	
Week ending Oct. 22, 1892	165	198,300	141,900	21,473	159.11	4 Boilers running 66 ins. Diameter. 18 ft. long. 55 4-in. Tubes.
" " " 29, "	168	200,140	143,400	21,152	158.78	
" " Nov. 5, "	121	142,730	103,500	16,703	160.70	
Week ending March 4, 1893	168	187,450	141,000	21,014	162.51	3 Boilers running
" " " 11, "	167	191,270	143,100	20,818	161.25	
" " " 18, "	163.30	183,400	138,600	20,921	161.19	

THE PRICES PAID FOR THE THREE ENGINES illustrate the recent advances in machine construction and design very forcibly. The 16,000,000 compound, in 1873, cost, with its boilers, $165,000, or $10,000 for each 1,000,000 capacity, and the guaranteed duty was 60,000,000 foot-pounds.

* See Mr. McMillan's statement, Appendix A.

The 12,000,000 compound, put in in 1881, cost $65,000 without boilers, or $5,400 per 1,000,000 gallons; the guaranteed duty being 97,000,000 foot-pounds. In 1890 the triple-expansion engine was put in for $76,000, including boilers, or $4,400 per 1,000,000 gallons, or, exclusive of the boilers, $66,000, equal to $3,666 per 1,000,000 gallons capacity.

Thus the 18,000,000 engine of 1893 cost practically the same as the 12,000,000 of 1881, with 29% better guaranteed duty; and the compound 16,000,000 engine of 1873 cost more than twice as much, with one-half the guaranteed duty of the 18,000,000 triple-expansion of 1893. These figures are the more interesting from the fact that all three of these engines are by the same builders.

COSTS OF OPERATION of the plant are exhibited in the following figures, compiled from data furnished by George H. Benzenberg, City Engineer, and Mr. McMillan, Chief Engineer of the city pumping stations.

The triple-expansion engine was put into regular operation in November, 1891, but was not in constant operation during the first part of 1892, owing to some apprehension as to the effect of fine anchor ice on the small valves; so that the engine was in operation, during the year 1892, a little over 290 days of 24 hours each. It has since been found that ice causes no more trouble with these valves than with the larger ones on the other engines, and, during the winter of '92–'93, this engine was kept in practically constant operation, the others being run to supply the amount of water required beyond the capacity of the triple engine. During the year 1892 this engine ran 6,980 hours, pumping 5,204,368,000 gallons of water, or about five-eighths of the total amount of water supplied to the city. The water was pumped against an average head of 160 feet, and the coal burned (anthracite) amounted to 5,516,000 pounds, costing $5.95 per ton of 2,000 pounds.

As there are three engines in this building it is not possible to exactly separate the cost of building and operation of plant chargeable to this particular engine; but Mr. Benzenberg's figures are as follows:

Cost of engine and four boilers, including ordinary engine foundation and the boiler settings	$81,000 00
Cost of so much of the building and boiler house as is chargeable to the engine	31,500 00
Total cost of plant	$112,500 00

COST OF OPERATING, 1892.

Coal, 2,758 tons at $5.95 per ton	$16,410 00
Engineers' wages (actual, $8,618), but if engine was in separate plant, would amount to	10,680 00
Repairs to engine and boilers	414 00
Oil and waste	372 60
Packing, etc	230 50
Total actual operating expenses	$28,107 10
Interest on plant at 5%	$5,612 00
Depreciation of plant at 4%	4,500 00
Total annual cost of operation and maintenance	$38,219 00

Dividing the above figures by the number of million gallons of water pumped (5,204,000,000), and we have as follows:

COST OF PUMPING 1,000,000 GALLONS FOR THE YEAR 1892.

	160 ft. high.	1 ft. high.
Cost, including fuel, labor, repairs, interest, and depreciation	$7 34	$4\frac{58}{100}$ cents.
Cost as above, but exclusive of interest and depreciation	5 40	$3\frac{37}{100}$ cents.
Cost per 1,000,000 gallons, for coal only	3 15	$1\frac{96}{100}$ cents.

The figures are for all coal burned, for all purposes, during the year, including starting and banking fires, heating the building, furnishing steam to a small engine which is in constant operation during the day driving the Water-works machine shop, and at night driving a dynamo, lighting the building.

The pumping station being located near the finest residence portion of the city, anthracite coal has always been burned, increasing the cost of pumping very materially. The matter of smoke-consuming furnaces is now under consideration, and if a suitable device can be found it is probable that bituminous coal will be ultimately adopted for use at this station.

The Sibley College test was made with Pocahontas bituminous coal at $4.65 per ton, or $1.20 less per ton, and doing about 10% better work than the anthracite ordinarily burned. Assuming, however, that this bituminous coal was only equal in calorific value to the anthracite, the difference in price per ton would make the cost of pumping 1,000,000 gallons as follows:

	160 ft. high.	1 ft. high.
Cost, including fuel, labor, repairs, oil waste, interest, and depreciation	$6 66	$4\frac{16}{100}$ cents.
Cost as above, but exclusive of interest and depreciation	4 72	$2\frac{95}{100}$ cents.
Cost per 1,000,000 gallons, for coal only	2 46	$1\frac{53}{100}$ cents.

These figures show a saving of 20% in cost of coal, and about 9% in total cost of pumping. At the 14th Street Pumping Station, in Chicago, where there are three engines of similar type, the cost of coal has been reduced from $2.16 per 1,000,000 gallons raised 100 feet high, with anthracite, to 60 cents per 1,000,000, with bituminous screenings, and to $1.24 with Shawnee bituminous coal. These figures are cited to show the possibilities for economy in the selection of cheaper grades of coal, where the conditions are such as to admit of its use.

THE PREPARATIONS FOR AND METHOD OF TEST were as nearly as practicable in accordance with the plan indorsed by a committee of the American Society of Mechanical Engineers, and as follows: *

The arrangement of the plant for test is shown in Fig. 97. Indicator diagrams were taken every fifteen minutes. Temperatures of feed-water, discharge jacket-water, external air, engine-room, and flue-gases were taken. The total consumption of water is obtained by measurement of the feed water supplied the boilers. This was taken from the hot well of another engine, and the condensed steam from the engine under test was, during the trial, discharged into the overflow and wasted. The steam used to supply the jacket was measured by receiving the overflow from the jacket-traps into weighing tank No. 3, and was returned to the boilers as a portion of the feed-water.

During the test two observers, one of whom was supplied by the builders of the engine, the other by the party from Cornell University, kept independent logs of the water supplied to the boilers.

The log gives the feed-water without correction; but in obtaining the full amount of water used in the jackets of the engines, a correction was necessary, for the reason that the discharge of water from the jacket-traps was at a temperature due to the pressure of steam on the jackets. This, being in excess of 212° Fahr., produced some loss by evaporation. Some of this small loss was prevented by passing the pipe from the jacket-traps through a large body of cold water. But a careful test was made to determine the amount of this evaporation, which indicated that, had no correction been made, the total amount of jacket steam would have been given as about 8% of the whole; but with the correction the jacket water becomes 9.25%.

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* *Sibley Journal of Engineering;* June, 1893, p. 302.

The fuel record was checked by independent observers, one of whom was furnished by the engine builders, the other by the college. The arrangement for measuring the temperature of the flue consisted in inserting a thermometer, the accuracy of which had been tested, and which could read to 700° Fahr., in the upper portion of the breeching, and by inserting a pyrometer into the center of the breeching beneath the thermometer. The pyrometer was a "thalpotasimeter." The quality of the steam, measured by a Carpenter throttling calorimeter, was very uniform, 98.95% throughout the test.

The flue-gas was obtained from time to time during the test and analyzed immediately; the apparatus used being a portable form of Orsat's apparatus.*

THE COAL burned was "Pocahontas," a soft, black coal, uniform in character, and very high in fixed carbon, and possessed wonderfully little ash. The coal was damp, and, on drying forty-eight hours over the boilers, a reduction in weight of 5.2% occurred. This coal showed 3.62% of moisture after reaching Sibley College. During the test the quantity of ash under the grates was less than 1%. The analysis showed 1.47% of ash; the discrepancy probably being due to ash drawn into the chimney by the draught; the total difference being less than 80 lbs. for the whole test. The analysis of the coal, as usually reported, gives: fixed carbon, 78%; hydro carbons, 20%; ash, 2%. A coal calorimeter gave results thus: Trial No. 1, 14,276 B. T. U.; Trial No. 2, 14,182 B. T. U.; Average, 14,232 B. T. U. The evaporation per pound of dry coal was 9.4 lbs. under the actual conditions of the test, which was equivalent to an evaporation of 10.72 lbs. from and at 212° Fahr. This is a low evaporation, considering the character of the coal, and no doubt largely accounted for by the small amount of coal consumed per square foot of grate, this being 7.15 lbs., equivalent to 6.76 lbs. of dry coal.†

THE INSTRUMENTS used for "indicating" the engines were care-

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* See Carpenter's *Experimental Engineering*, or Thurston's *Engine and Boiler Trials*, for details of methods of work.

† It is important to note that, notwithstanding the extraordinary character of this fuel, the evaporation was not at all exceptional, and the engine gained nothing by its use over what should have been anticipated with any really good coal. This singular fact is attributed partly to too light fires, and partly to the fact that the hydro-carbons seem rarely to give their full effect, not to a defect of the fuel in any peculiar characteristic.

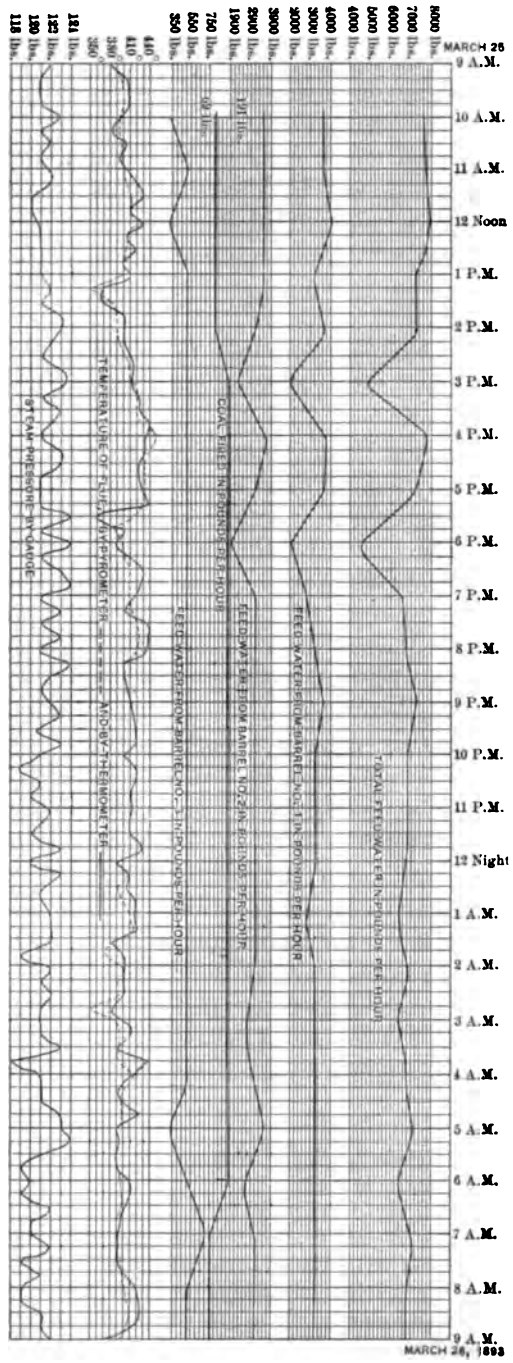


FIG. 98.—GRAPHICAL LOG.

fully calibrated, the scale of the springs used for the high-pressure being 60 lbs., for the intermediate-pressure 20 lbs., and for the low-pressure 16 lbs. These springs were calibrated under the conditions of actual use and in the indicators.

The dimensions of the engines and pumps were obtained by Mr. G. H. Benzenberg, City Engineer. His certificate was appended to the report. Before commencing the test the pumps were carefully examined by Mr. Hamilton, the first engineer of the station, and the professor in charge, and found perfectly tight.

THE SIBLEY COLLEGE TRIALS were inaugurated March 23, by a preliminary organization of the crew and a practice test of several hours' duration, in which the observers were given a drill in their assigned work; the engine was examined and inspected in every part, and the whole system of apparatus was tested, to ascertain its fitness for the purposes in view.

All arrangements being found satisfactory, the final trial was proceeded with, March 25, at 9 A. M.; was continued uninterruptedly twenty-four hours; was commenced and ended to the satisfaction of all four sets of observers, and was so certified by the representatives of the city of Milwaukee, of the builders, of the delegate from R. W. Hunt & Co., and by the official representative of Sibley College.

It was not necessary to clean the fires, or even to touch them with rake or slice-bar, *during the whole test;* and they were cleaned only after the completion of the test. Flue-gas analyses were made several times; the results in every case showing excess of air and no CO. The temperature of the flue was about 56° higher than that of the steam, 403° Fahr. as read by the pyrometer, and 406° Fahr. as read by the thermometer.

The weight of water delivered was computed from the plunger displacement, and the total weight for each revolution of the engine being 5,229.29 lbs.* The average head pumped against was 161.84 feet. The head was measured by a pressure gauge connected to the discharge chambers of the pumps.

* This is a standard method, and was, in fact, the only method practicable at the time. The writer would always check by the use of a weir, when possible; but, in this case, the construction of the pumps was such as to make sensible slip improbable, and the pump-cards are so sharp at the corners as to indicate that they ran full. The writer made a visit to Milwaukee and personally examined the engine, to satisfy himself of the fact that the valves act promptly and efficiently. Experiment, later, on an engine of similar construction, gave a slip of 0.0073. It may be taken here as undoubtedly less than one per cent.

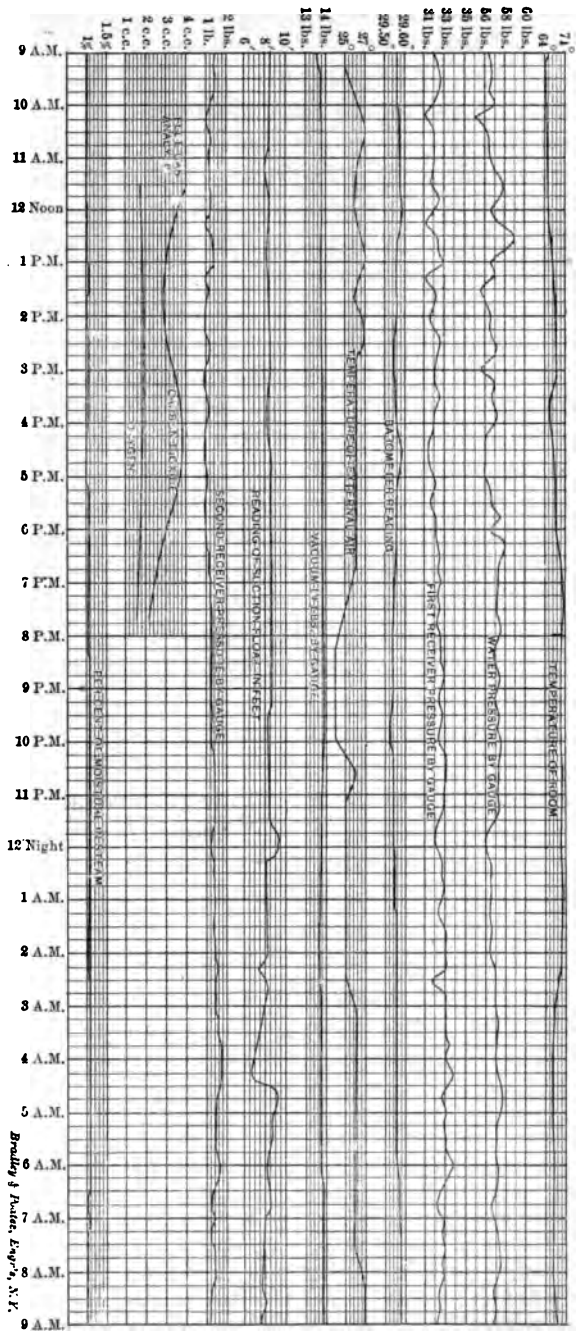


FIG. 99.—GRAPHICAL LOG.

The total head increased slowly during the day, due to a rise in the reservoir; the average for the watches being:

First watch, 9 A. M. until 2 P. M.	161.46
Second watch, 2 P. M. until 10 P. M.	161.644
Third watch, 10 P. M. until 6 A. M.	162.008
Fourth watch, 6 A. M. to 9 A. M.	162.365

THE DATA, LOGS, AND RESULTS OF THIS TRIAL are given in Appendix B, and the details of the work in the report made to the builders.* They are admirably summarized in the accompanying "graphical log." (Figs. 98 and 99.)

As there shown, the indicated horse-power was 573.87, divided among the cylinders thus: High-pressure, 175.39; intermediate, 169.62; low-pressure, 228.86. The steam pressures were, at the high-pressure cylinder, 121.46; first receiver, 32.43; second receiver, 1.5. The total water evaporated into dry steam was 6,700.7 lbs. per hour, *equivalent to* **11.678** *lbs. per I.H.P. per hour.*

The heat supplied per minute averaged 124,890.7 B. T. U., equivalent to 217.6 B. T. U. per I.H.P., which includes heat carried over by entrained water. The useful work computed from the pressures and volume of water delivered was 520.96 D.H.P. The friction of the engine was 52.92 H.P., 9.22% of the I.H.P. The work was 1,031,520,000 foot-pounds per hour; the coal consumed was 719.8 lbs.; giving a duty per 100 lbs. of coal, of 143,506,470 foot-pounds, or for 1,000 lbs. of feed-water, 152,448,000 foot-pounds. This duty, on the basis of 1,000,000 B. T. U. supplied, 137,056,000 foot-pounds.

The quality of steam was:

High-pressure cylinder, at cut-off	89 %
" " " release	93 %
Intermediate, at cut-off	86.0 %
" release	97.5 %
Low-pressure, at cut-off	77.8 %
" " release	85.6 %

A SPECIAL TEST, made April 7, 1893, determined the heat rejected into the condenser, resulting as below:

Average revolutions per minute	20.41
Average steam-pressure gauge	121.6
First receiver pressure	31.9
Second " "	1.67
Vacuum below atmosphere	13.5

* *Sibley Journal of Engineering*, June, 1893.

Temperature of injector-water	40°
" discharge-water	98°
Depth over weir, 12-inch notch, feet	0.2669
Cubic feet per minute	26.78
Pounds of injection-water per minute	1,660
Pounds of feed-water per minute	111.6
Ratio injection-water to feed-water	14.9
Per cent. of heat carried off in jacket-water	9.25
" " " " exhaust	75.10
" " " " discharge water	15.65
Probable quality of steam in exhaust from low-press. engine.	84.3%

Figs. 100, 101, 102 show the indicator diagrams obtained from engines and from pumps.

THE NET RESULT OF THE WHOLE INVESTIGATION is seen to be the establishment, by standard methods of contemporary practice, of the highest record of steam-engine duty yet obtained as the outcome of such exact methods of research. The useful work performed, measured as "duty" by the accepted standards, is seen to be, as computed by Professor Carpenter's staff:

Foot-pounds of work for 100 lbs. dry coal	143,306,470
" " " " " wet "	135,770,000
" " " " " combustible	145,438,000
" " " 1,000 lbs. dry steam	154,048,000
" " " 1,000 lbs. feed-water	152,448,000
" " " 1,000,000 B. T. U.	137,656,000
" " " 1 cwt. coal (112 lbs.)	152,630,000
Kilogrammeters of work per kilo of coal	429,110

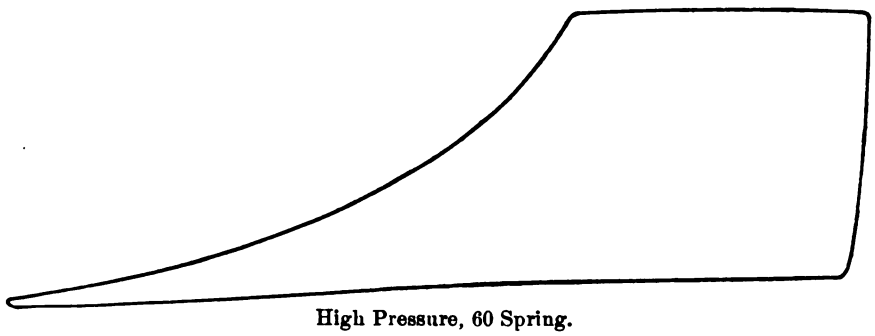
While the thermo-dynamic results are:

Thermo-dynamic efficiency	0.194
Heat per *I. H. P.* per hour, and per minute, *B.T.U.*..13,056 ;	217.6
Steam per *I. H. P.* per hour, lbs.	11.678
Fuel per *I. H. P.* per hour, lbs.	1.237
Heat per *D. H. P.* per hour, and per minute, *B. T. U.*14,382 ;	239.7
Steam per *D. H. P.* per hour, lbs.	12.864
Fuel per *D. H. P.* per hour, lbs.	1.364

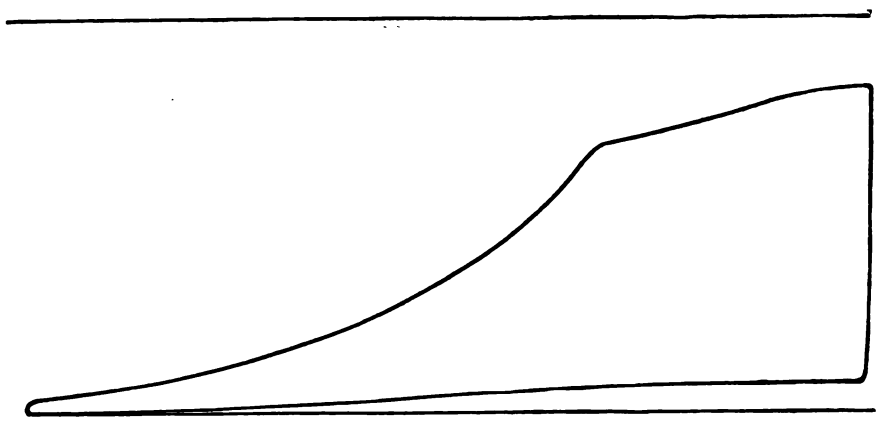
Were these engines attached to boilers evaporating 10 lbs. instead of 9, the fuel-rate would be, per *I. H.P.*, 1.168 lbs., and per *D. H. P.*, 1.286, or about $1\frac{1}{6}$ and $1\frac{1}{4}$ lbs., respectively. The evaporation actually observed during the trials, was, as seen on the log, but 8.81 ; the apparent evaporation only 8.9, and the equivalent, "from and at 212," but 10.27 ; while the same figures for the coal, exclusive of its moisture, were, respectively, for apparent evaporation, 9.425 ; and, for equivalent, from and at the atmospheric boil-

ing-point, 10.72. Per pound of combustible, they become a tenth of a pound higher.

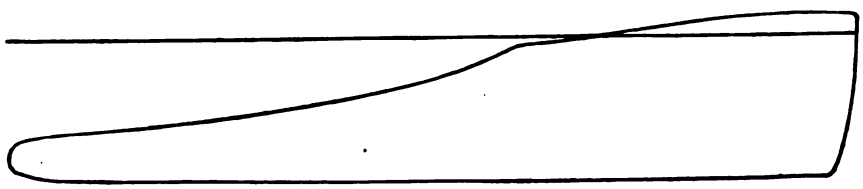
The above-given efficiency is 0.668, or two-thirds, that of a



High Pressure, 60 Spring.



Intermediate, 20 Spring.



Low Pressure, 16 Spring.

FIG. 100.—INDICATOR DIAGRAMS, TOP END.

Carnot cycle working through the same range of temperatures and pressures. It is equal to $0.1873 \div 0.252 = 0.74$ of the thermodynamic efficiency for the Rankine cycle of the ideal case. It

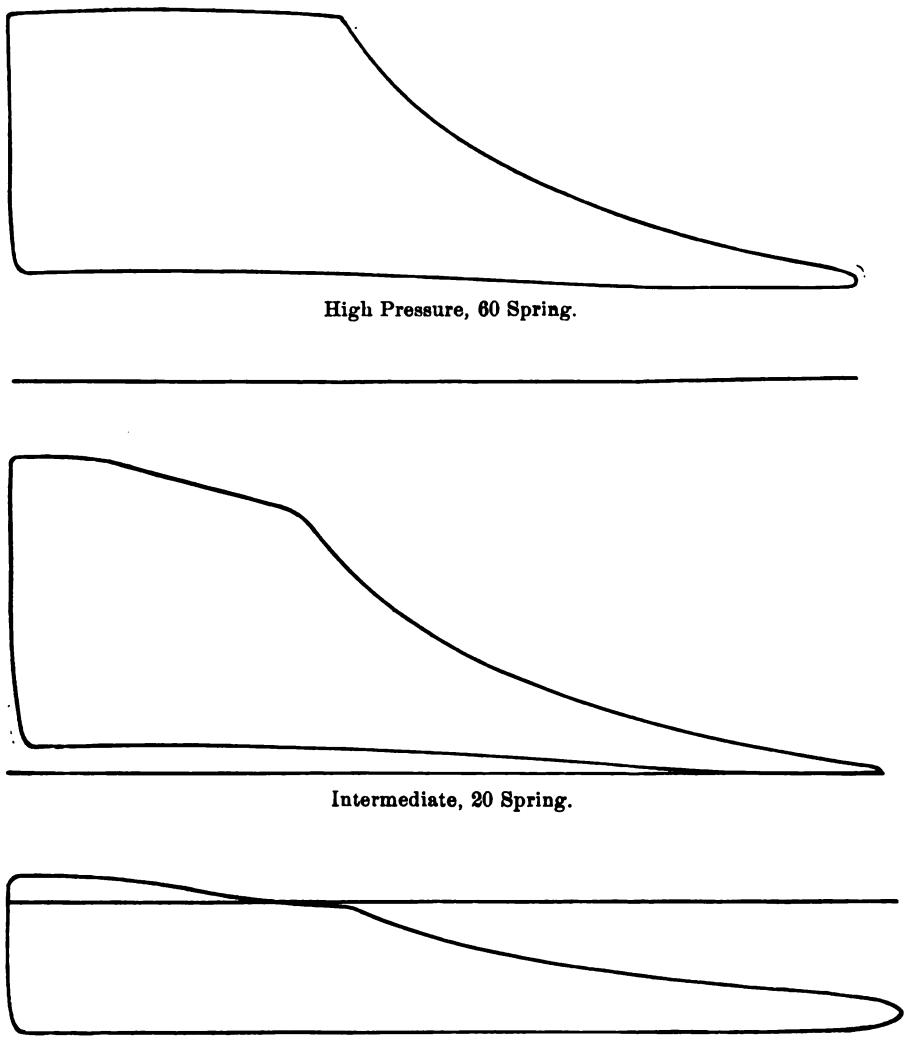


FIG. 101.—INDICATOR DIAGRAMS, BOTTOM END.

corresponds very exactly with the previously computed final efficiency, including the estimated wastes; which computation was made some time before the trial, to obtain some idea of what was to be expected.

We will now endeavor to ascertain the nature and extent of the wastes in this engine and the secret of its remarkable efficiency.

THE WASTES OF THE STEAM-ENGINE are: (1) the thermo-dynamic and unavoidable wastes due to the fact that, at the minimum limit of pressure, in the operation of the engine, a proportion of the heat-energy supplied, always considerable, and precisely determined by that limit, must inevitably be rejected from the system and lost to all thermo-dynamic use; (2) the wastes due to

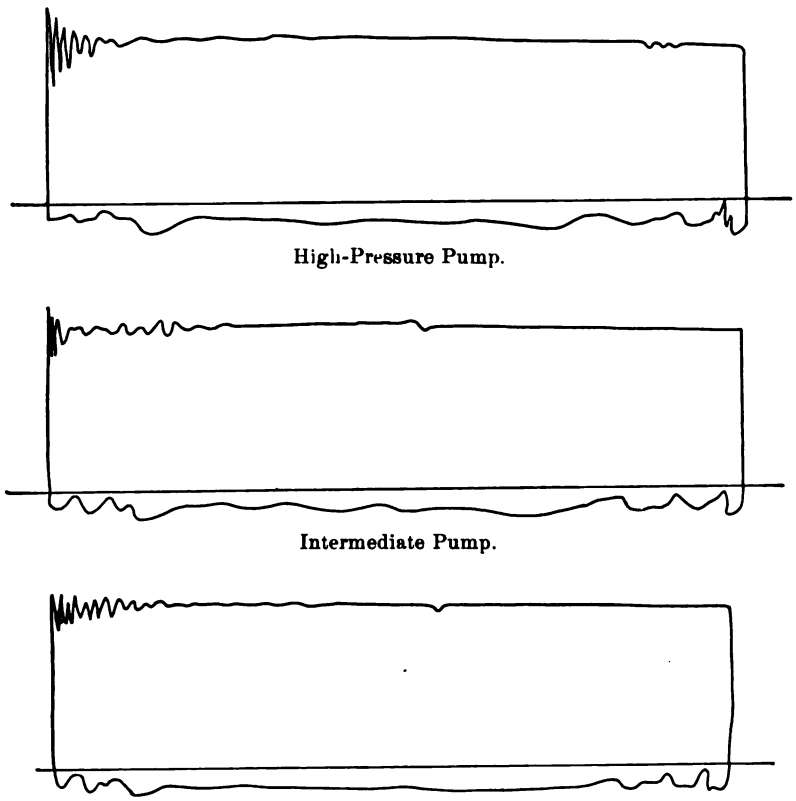


FIG. 102.—INDICATOR DIAGRAMS, PUMPS.

the conduction and radiation of heat from the exterior of the machine to surrounding bodies, with similar result; (3) the internal wastes of conduction and transfer from steam to exhaust side without transformation into work; (4) the wastes of dynamic energy by friction of the various rubbing surfaces of the engine. Of these the first is beyond remedy except by such reduction as may be possible by extension of the expansion to the limit of a ter-

minal pressure coinciding with the back-pressure; the second is controllable and largely reducible by suitably clothing the exterior of the engine with non-conducting materials; the third is also reducible by reduction of the time permitted for the waste to take place and by making the working fluid and the surfaces exchanging heat with it of minimum conductivity and heat-storing power; the fourth waste is only capable of reduction by the amount to which the friction of the machine may be reduced to minimum proportions as compared with the gross power developed. The first quantity has been brought down from an original value of 90% to 85, in many cases, possibly to 75 in some instances; the second from 10, perhaps even 20, to 5 frequently, sometimes to less; the third from 30 or 50%, to, as in the present case, about 10%; and the last waste has come down from 25 to 10%, though usually in condensing engines ranging from 12 to 15.

Could steam be worked in the Carnot cycle, entering at maximum pressure, expanding adiabatically to minimum, rejecting heat with compression during the return-stroke until adiabatic compression could be made to restore it to its initial pressure and volume, thus converting only latent heat of expansion into work, and that to the full extent of the Carnot law $E = (T_1 - T_2) \div T_1$ there would be required only the following quantities per horse-power per hour:

STEAM DEMANDED; IDEAL CASE.

Engine condensing; $p_b = 1.5$; Complete Carnot Cycle.

Steam-pressure (abs.).....	45	75	115	165	215	315
Efficiency.....	0.215	0.248	0.278	0.302	0.320	0.347
B.T.U. per horse-power per hour	11,838	10,221	9,119	8,427	7,938	7,335
Steam " " "	12.8	11.4	10.5	9.8	9.5	9.0
Fuel " " "	1.28	1.14	1.05	0.98	0.95	0.90

It is seen that the engine here to be studied, operating at a pressure of 125 lbs., as originally designed, would demand, if it were possible to work through this cycle, about 10 lbs. of steam per horse-power per hour, or about 1 lb. of fuel; the evaporation being taken, as it might fairly be for this case and cycle, at 10 lbs. of steam per pound of good fuel. The difference between this figure and that computed for the somewhat different cycle actually and necessarily adopted, measures a waste due to imperfection of the thermo-dynamic cycle employed—a waste

not included in the above summary or considered hereafter. In the cycle to be studied, that of the engines, ideal and real, to be here compared, it is impossible to secure that adiabatic compression which is required to bring the compressed working fluid back to initial pressure before the succeeding cycle is begun. This is the characteristic difference between it and the ideal Carnot cycle, usually taken as standard in thermo-dynamic studies.

The ideal cycle taken as representative of the real cycle of the steam-engine, as customarily built and operated, assumes admission at boiler-pressure, adiabatic expansion to back-pressure, and then complete rejection of the charge from the system. The heat expended is seen to be, not the latent heat of vaporization, simply, but the sum of that quantity and the sensible heat required to raise the temperature of the feed-water from that at which it is supplied to the boiler, to that at which it is converted into steam. The difference between this and the preceding Carnot cycle, and which is wasted in this case, even in the ideal engine, is the difference between the "total heat of evaporation" from the feed-water temperature and at the temperature of the boiler-steam, and the latent heat of vaporization, at the latter temperature, nearly. These quantities, for the specified pressures, are as follows:

HEAT-ENERGY OF STEAM.

Pressure (absolute). *P.*	"Total heat" (from 110° F.). *T.*	Latent heat of evaporation. *L.*	Sensible heat. *S.*	Ratio of *S/L.*	Ratio of *S T.*
30	1080	939	141	0.15	0.13
60	1093	909	184	.20	.17
100	1104	884	220	.25	.20
125	1109	872	237	.27	.21
150	1113	862	251	.29	.23
200	1120	845	275	.32	.25
300	1131	818	313	.38	.28

The wastes of the representative cycle of the real engine are thus exaggerated, as compared with the cycle of best effect, in consequence of the practical impossibility, in our engines, as built, of raising the working fluid, then a mixture of steam and water, from the final pressure and temperature to the pressure and temperature of the initial stage, by mechanical compression; and this loss amounts to about 20% of the total heat of the steam, or to about 25% of the latent heat operating thermo-dynamically in

the Carnot cycle, for pressures of 100 lbs., absolute, and a little over. It becomes an increasing percentage of waste as pressures rise, and, at 300 lbs. pressure, the Carnot cycle becomes nearly 40% more efficient than the representative ideal; the latter wasting in this manner about 30% of all the heat supplied and 40% of the quantity supplied the former. This measures the importance of finding a method of successfully employing the Carnot cycle in the steam-engine, a plan often sought, but, as yet, never found.

THE HEAT STORED IN STEAM, in the operation of the steam-boiler under the customary conditions, consists of the sum of the sensible and latent heats absorbed by the fluid, in its conversion from water at the temperature of the feed, into steam at the temperature and pressure of its utilization. The quantity stored per unit of weight exceeds that which would be stored in the Carnot process, as the total heat, for the actual case, exceeds the latent heat of vaporization at the proposed working pressure. The higher the temperature of the feed-water, the less the heat thus stored in the unit of fluid passing through the cycle. It is a minimum in the Carnot cycle, a maximum when the feed-water receives no heat from the fluid rejected from the engine-cylinder. All common and good practice lies between these extremes, approaching the former as the feed-water heating, by application of the rejected heat of the engine, is the more effective.

The measure of this stored energy, passing into the steam at the boiler, is known to be very nearly

$$t = 1082 + 0.305\,(t_2 - t_f),\ (B.T.U.)$$
$$t_m = 606.5 + (t_{2m} - t_{fm}),\ (\text{calories});$$

where t and t_m represent the total heat in British and metric measures, respectively, t_2 and t_f being the temperatures of the steam and of the feed-water in the same measures. In the Carnot cycle, previously discussed, $t_f = t_2$, and the latent heat, which, only, is supplied, is

$$l = 1114 - 0.695\,t_2;$$
$$l_m = 606.5 - 0.695\,t_{2m}.$$

In the common cycle, more heat is stored per unit of weight; but in the Carnot cycle a larger proportion is utilized; in the former, a smaller number of pounds of steam per horse-power is required for efficiency unity; but in the latter less heat per horse-power for a given cycle or range of temperature worked through.

The energy stored by transfer from furnace to steam is given in the succeeding table, assuming a feed-temperature of 140° Fahr. (60° C.), for a number of pressures, up to 500 lbs. by gauge:

ENERGY STORED IN STEAM.

Pressure.		Thermal Units.		Foot-pounds.
Absolute.	By gauge.	British.	Metric.	
515	500	1117.42	279	869,352
250	235	1096.21	274	853,400
150	135	1083.2	271	843,100
100	85	1073.86	268	835,800
60	45	1063.17	266	827,800

The total energy stored in steam at the boiler under usual conditions of temperature of feed-water, and of temperature and pressure of steam supply, seldom varies greatly from 1,000 *B. T. U.*, the minimum being about 950 for low-pressure heating boilers, with water returned at nearly the boiling-point; the maximum figure being that of non-condensing engines at 100 lbs. pressure, by gauge, the feed-water being taken from the neighboring streams or springs, and in the case of the later forms of high-expansion engines, supplied with steam at 180 or 200 lbs. pressure, and feed-water from condensers held at 110° to 120° Fahr., in which latter cases the supply is not far from 1,100 *B. T. U.* per pound of the fluid. In exceptional cases of lower temperatures of feed, this figure may rise to 1,200 *B. T. U.* Measured in dynamic units, this energy thus amounts to from 750,000 to 1,000,000 foot-pounds; and the consumption of steam, for efficiency unity, would thus be, per hour, from 2.5 to 2 lbs. per horse-power. It is taken in the examples which are given in illustration of the computations of test efficiencies at 2.3; but this quantity, in all cases in which great accuracy is desired, should be computed for the conditions, as to feed-water and steam temperatures, which distinguish the case. These computations may be made by reference to the tables of steam-pressures and temperatures, in all of which the amount of heat-energy stored is given, as measured from some standard temperature, as the melting point of ice or the zero of the temperature scale. Taking the difference between the heat thus expressed for the feed-water, and that given for the steam at

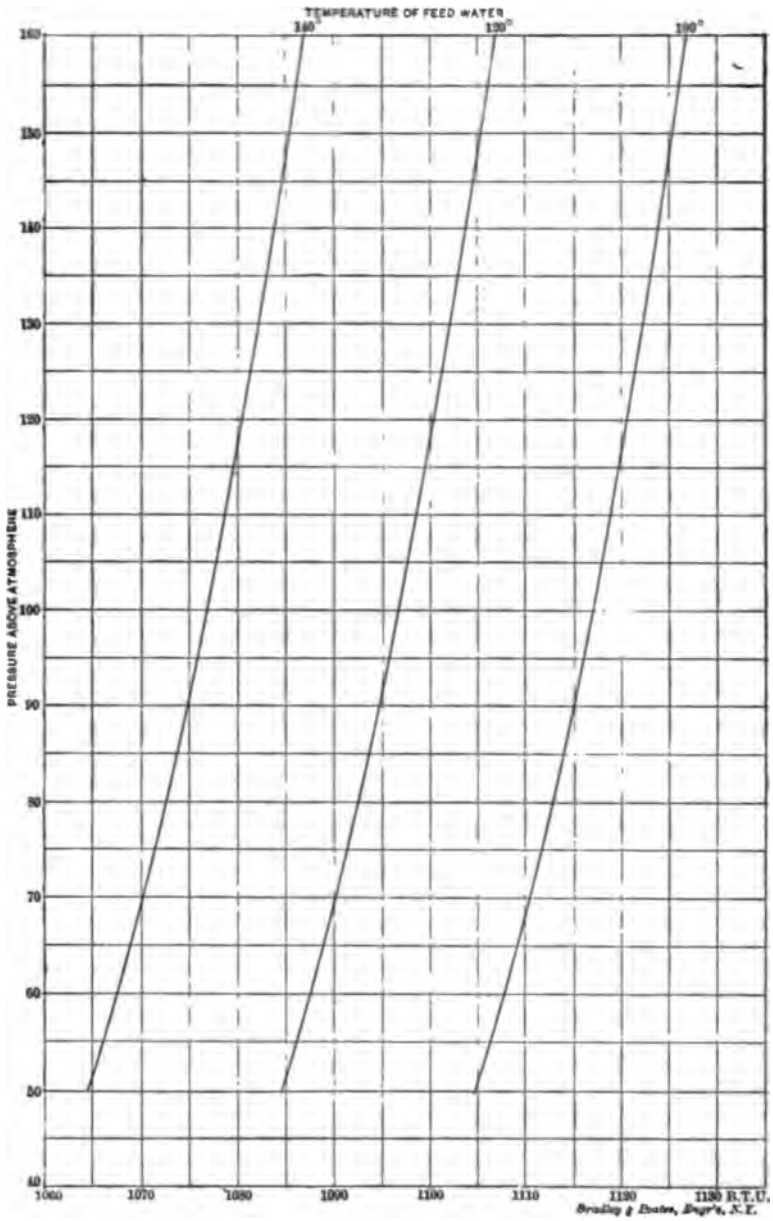


FIG. 103.—ENERGY STORED IN MAKING STEAM.

the stated temperature, this difference, multiplied by 772 or 778, gives the required quantity.

In illustration of the process: assume boiler pressure 120 lbs. per square inch above the atmosphere, 135, nearly, absolute, and feed-water to enter at 110° Fahr. Then we have

$$p_1 = 120 + 14.7 = 134.7.$$
$$H_1 = 1{,}188.42\ B.\ T.\ U.\ \text{(from steam table).}$$
$$f_4 = 110°\ \text{Fahr.}$$
$$H_1 - (110° - 32) = 1{,}110.64\ B.\ T.\ U.$$
$$= 864{,}079.48\ \text{foot-pounds.}$$

Since 1 H. P. = 33,000 foot-pounds per minute, or 1,980,000 per hour, it is equivalent to

$$\frac{1{,}980{,}000}{778} = 2{,}545\ B.\ T.\ U.\ \text{per hour.}$$

$$\frac{33{,}000}{778} = 42.42\ B.\ T.\ U.\ \text{per minute.}$$

Thence, for efficiency unity, this is

$$\frac{2{,}545}{1{,}110.64} = 2.3$$

pounds of steam required per horse-power per hour.

Were this steam worked in a Carnot cycle between the stated temperatures, the efficiency would be

$$Eff. = \frac{811 - 571}{811} = 29.6\%,$$

and the energy available in the heat supplied would be

$$1{,}110.64 \times 0.296 = 328.8\ B.\ T.\ U.$$

per pound of steam supplied under such circumstances with the stated quantity of heat. Each pound would, however, in this case, only accept a supply measured by its latent heat of evaporation. The corresponding *B. T. U.* and pounds of steam per horse-power per hour would be

$$2545 \div .296 = 8598\ B.\ T.\ U.;\ \frac{2.93}{.296} = 9.9\ \text{lbs.},$$

and the steam used would be increased about 20% by the fact just mentioned, if worked in a reversible cycle like that of Carnot.

In the first case, for other efficiencies than unity, we have

$$Eff. = \frac{2.3}{W.};$$

where W is the weight of steam per horse-power, and in any case the efficiency will be

$$Eff. = \frac{2{,}545}{B.\ T.\ U.};\ \text{or}\ Eff. = \frac{42.42}{B.\ T.\ U.};$$

where the denominator is, in the one expression, the heat demanded per horse-power per hour, and, in the other, that supplied per minute.

In the Carnot cycle the feed-water circulates between boiler and engine, entering the former at the temperature due the working pressure; the "feed-water" being heated by adiabatic compression, and only latent heat supplied by the furnace.

It will be noted that heating the feed-water entering the boiler, decreasing the range of temperature between that of water supply and that of steam delivery, decreases the maximum thermo-dynamic efficiency of the corresponding Carnot cycle, reduces the quantity of heat stored in the unit mass, and increases both the weight of steam storing unity of heat and the quantity of steam of which the energy thus stored corresponds to the measure of a horse-power. The actual gain by heating feed-water comes of the defect in the operation of the steam-engine cycle, resulting in failure to raise the fluid from minimum to maximum temperature by compression in the working cylinder, and compelling the supply, at the boiler, of heat equivalent to this waste, together with that additional quantity needed to raise the water from the temperature of its source to the minimum limit of the engine cycle. Where this can be obtained from otherwise wasted heat, the saving may prove very important.

In this case, the lower limit is fixed by extra thermo-dynamic conditions, and, at the boiler, is always below that at the engine-cylinder, and its variations determine corresponding variations of the quantity and cost of the heat supply.

Computing the energy stored in making steam at a number of

usual absolute pressures, measured in B. T. U. and in foot-pounds, we obtain the following tabulated quantities:

ENERGY STORED IN ONE POUND OF STEAM IN B. T. U.

Pressures in lbs. per square inch.	Feed-Water Temperatures.					
	110	130	150	170	190	210
90	1,101.569	1,081.569	1,061.569	1,041.569	1,021.569	1,001.569
100	1,103.866	1,083.866	1,063.866	1,043.866	1,023.866	1,003.866
110	1,105.986	1,085.986	1,065.986	1,045.986	1,025.986	1,005.986
120	1,107.961	1,087.961	1,067.961	1,047.961	1,027.961	1,007.961
130	1,109.809	1,089.809	1,069.809	1,049.809	1,029.809	1,009.809
140	1,111.555	1,091.555	1.071.555	1,051.555	1,031.555	1,011.555

ENERGY STORED IN ONE POUND OF STEAM, IN FOOT-LBS.

Pressures in lbs. per square inch.	Feed-Water Temperatures.					
	110	130	150	170	190	210
90	857,021	841,461	825,901	810,341	794,781	779,321
100	858,808	843,248	827,688	812,128	796,568	781,008
110	860,457	844,897	829,337	813.777	798,217	782,657
120	861,994	846,434	830,874	815,314	799,754	784,194
130	863,431	847,871	832,311	816,751	801,191	785,631
140	864,790	849,230	833,670	818,110	802,550	786,990

Assuming these energies to be applied in thermo-dynamic operations in Carnot cycles of similar total temperature range, we should obtain the following tabulated efficiencies and quantities of energy utilized in units corresponding to unit weight of steam:

CARNOT EFFICIENCIES.

Pressures in lbs. per square inch.	Minimum Temperatures.					
	110	130	150	170	190	210
90	.270	.243	.218	.193	.167	.141
100	.276	.250	.225	.200	.174	.149
110	.282	.257	.232	.207	.182	.156
120	.288	.263	.238	.213	.188	.163
130	.294	.269	.244	.219	.194	.169
140	.299	.274	.249	.224	.200	.175

AVAILABLE ENERGY, IN B. T. U., PER POUND OF STEAM.

Pressures in lbs. per square inch.	Minimum Temperatures.					
	110	130	150	170	190	210
90	297.424	262.821	231.422	201.023	170.602	141.221
100	304.667	270.966	239.370	208.773	178.153	149.576
110	311.888	279.098	247.309	216.519	186.729	156.934
120	319.093	286.134	254.175	223.216	193.257	164.298
130	326.284	293.159	261.033	229.908	199.783	170.658
140	332.355	298.386	266.817	235.548	206.311	177.022

AVAILABLE ENERGY, IN FOOT-LBS., PER POUND OF STEAM.

Pressures in lbs. per square inch.	Minimum Temperatures.					
	110	130	150	170	190	210
90	228,096	204,475	180,046	156,396	132,728	109,870
100	236,591	210,812	186,208	162,425	138,603	116,370
110	242,649	217,417	192,406	168,452	145,275	122,021
120	248,254	222,612	197,748	173,662	150,354	127,824
130	253,849	228,078	203,084	178,868	155,431	132,772
140	258,572	232,144	207,584	183,256	160,510	137,729

STEAM PER I.H.P. PER HOUR AT MAXIMUM EFFICIENCIES, AS ABOVE.*

Pressures in lbs. per square inch.	Minimum Temperatures.					
	110	130	150	170	190	210
90	8.68	9.58	10.98	12.64	14.92	18.02
100	8.36	9.39	10.62	12.18	14.30	17.01
110	8.15	9.12	10.28	11 75	13.63	16.22
120	7.97	8.88	10.02	11.41	13.16	15.38
130	7.80	8.68	9.74	11.08	12.73	14.91
140	7.65	8.52	9.54	10.80	12.33	14.38

Computing by means of the familiar formulas the total heats stored in steam at various gauge pressures, and assuming the feed-water to enter at the boiling-point under atmospheric pressure, only so much of this heat is available as may be transformed into work by complete expansion to the lowest possible final pressure, which is, in the ideal case, the back pressure of the

* The table exhibits the *steam circulated* in the *engine*, and equals 1,980,000 foot-pounds, divided by the available energies above.

atmosphere in the non-condensing engine and that of the condenser in the condensing machine. In the case of initially dry steam it is the quantity

$$U = J\left[T_1 - T_2\left(1 - \log_e \frac{T_1}{T_2}\right)\right] + \frac{T_1 - T_2}{T_1} H_1;$$

it being assumed that expansion is complete. If, for example, the lower limit is the atmospheric pressure, this computed quantity becomes as in the following table, and illustrated in the accompanying figure, in which are given the curves exhibiting the variation of the energy of the sensible heat of the water and of latent heat of vaporization, with that of their sum, the total heat of the steam as measured as above.

The first term in the second member of the above equation measures the increased work of the unit weight of steam in the Rankine, as compared with that of the Carnot, cycle; while the increased cost of that work, as a minimum, is $J\ (T_1 - T_2)$, and the net result is seen in the decreased efficiency already adverted to.

AVAILABLE STORED ENERGY OF STEAM.

Selected pressures; minimum pressure and temperature those of the atmosphere and boiling-point corresponding.*

LIMIT OF WORK IN NON-CONDENSING ENGINE.

Pressure.		Foot-Pounds Energy Per Pound Steam.		
Absolute.	Gauge.	Water.	Lat. Heat.	Total.
40	25.3	1,645.7	54,111.7	55,757.4
60	45.3	3,449.2	74,884.6	78,333.8
80	65.3	5,206.5	89,138 7	94,345.2
100	85.3	6,885.2	99,787.6	106,673.8
120	105.3	8,483.1	108,325.4	116,808.5
140	135.3	9,992.6	115,382.1	125,374.7
160	145.3	11,444.2	121,323.6	132,767.8
180	165.3	12,821.4	126,499.1	129,320.5
200	185.3	14,153.3	130,967.4	145,120.7
220	205.3	15,463.1	136,003.3	151,466.4
250	235.3	17,314.4	140,516.0	157,830.4
500	485.3	30,055.5	165,892.7	195,948.2
1,000	985.3	48,671.5	179,212.0	227,883.5
5,000	4,985.3	130,494.2	183,305.9	313,800.1
10,000	9,985.3	193,787.0	145,376.8	339,164.2

* *Manual of Steam Boilers*, p. 286, p. 827, Table XI.; *Manual of the Steam Engine*, Part I.

Plotting the tabulated figures and determining the form of the curve representing the law of variation of each, we obtain the diagrams exhibited in the accompanying engraving, where in Fig. 12 are given the curves of available energy of the water, of latent heat, and of steam.* The curve of energy of latent heat is not only peculiar in its variation in radius of curvature, but also in the fact of presenting a maximum ordinate at an early point in its course. This maximum is found at a pressure of about one ton per square inch.

This curve is symmetrical about one of its abscissas, and it must have, if the formulas hold for such pressures, a point of contrary flexure at some enormously high pressure and temperature. The formula is not, however, a "rational" one, and it is by no means certain that the curve is of the character indicated.

It is interesting to observe that the available energy of sensible and latent heats follow entirely different laws, with rising pressures and temperatures of vaporization. While that of sensible heat—of heated water, as it is expressed on the diagram—steadily, and apparently continually, rises, that of latent heat rises rapidly throughout the range of familiar pressures, takes on a slower rate of acceleration after passing about 100 lbs., and, with decreasing rapidity of rise, finally ceases to increase, and passes a maximum at about one ton per square inch, thence falling off slowly and indefinitely. The curve of total energy, the resultant of these conspiring quantities, rises constantly and indefinitely, but its rate of increase diminishes observably after passing 100 lbs., and gradually approaches a constant value which it probably never reaches. It should, however, be remembered that our formulas are only known to be sensibly accurate for a comparatively low range of pressure and temperature, and the curves at hitherto unattained points may take quite different paths. The engineer is practically restricted, so far as can be at present seen, to pressures and temperatures within those at which his materials approach the red heat and begin to lose seriously in strength and safety from molecular flow, and this probably places the limit not higher than 800° Fahr. (427° C.), and probably not above 700° Fahr. (370° C.), or at about 333 or 200 atmospheres, about 5,000 or 3,000 lbs. per square inch, respectively. But the conclusion of most practical interest at the moment is that we may expect that, whatever the thermo-

* *Steam Boilers as Magazines of Explosive Energy*, by R. H. Thurston. *Trans. A. S. M. E.*, November, 1884, Vol. VI.

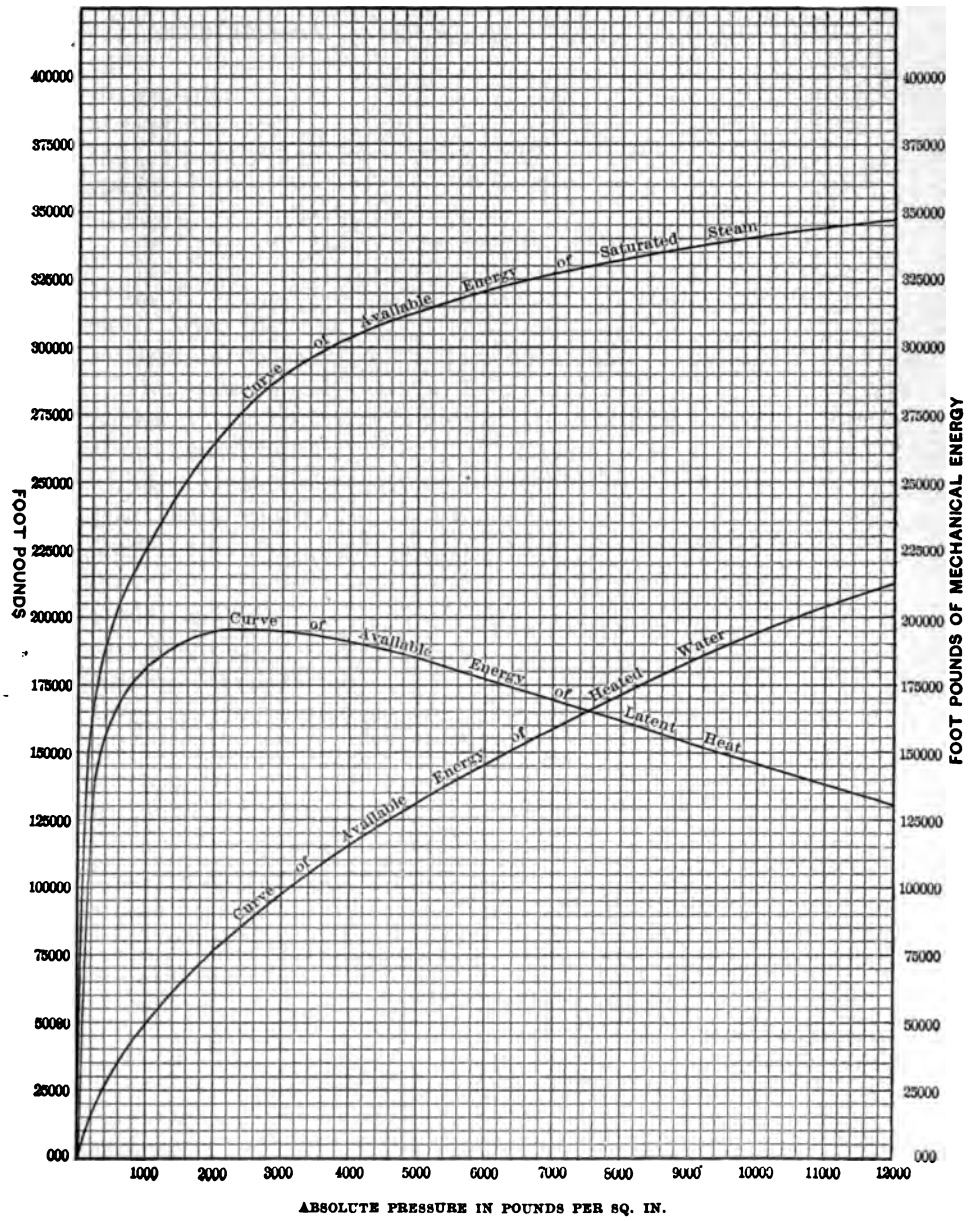


FIG. 104.—HEAT-ENERGY IN STEAM AND WATER.

dynamic advantage of increasing pressures hitherto experienced, it may be fairly anticipated that those advantages will continue to attend the increase of steam pressures for an indefinite period in

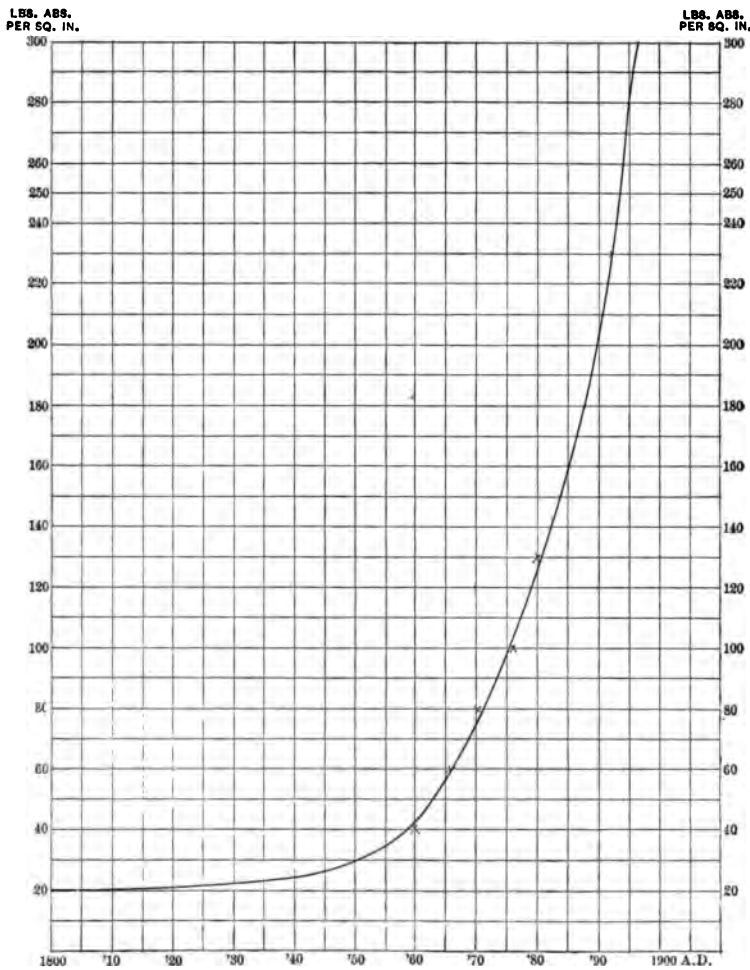


FIG. 105.—PROGRESS IN STEAM PRESSURES, 1800–1900.

the future. Progress will probably come, as hitherto, by our gain in ability to safely and profitably handle higher tensions.

In this connection it is interesting to observe the method of variation of pressures in practice during the century, as seen illustrated in the accompanying diagram (Fig. 105), which gives the

data obtained by the study of marine practice, as prepared for the writer by Mr. T. Hall. The rise of pressure has been, on the whole, a fairly uniform one in rate and regularity during the generation just past, and, if this should continue another generation, the indications are that the year A.D. 1900 may see 300 and possibly 400 lbs. in common use; and it would seem possible that only the approach to temperatures rendering our materials of construction no longer reliable may limit this advance. At the current rate of progress it will still be a century before we make standard those pressures which were experimentally employed, and with safety and comparative economy, by Jacob Perkins a half-century ago. Forty years hence, the promise of our diagram is that customary pressures may be as high as 400 lbs. per square inch—27 atmospheres. The sharp bend in the curve limits the period of the simple engine of James Watt and marks the inauguration of the period of multiple-expansion engines. The steadiness with which the curve adheres to its uniformity of radius subsequent to that period would ordinarily be taken as evidence that we may fairly expect steady and uniform progress at a similar rate in the immediate future; but no one can say at what moment obstacles of insurmountable, or of slowly surmountable, character may arise to change the law of the curve. Perkins, however, in 1824–36, employed pressures of, first, 500, then 1,000, and finally of about 1,500 lbs. per square inch, with a ratio of expansion of 8, and obtained, as he says, one horse-power with but two bushels of coal per day, or less than two pounds per horse-power per hour, with but a fraction of the expansive action practicable to-day. With modern sectional boilers, and skill in design and construction, and guided by the light of modern science, now illuminating a then dark field so completely, it would be strange if our engine builders could not do better than Perkins, and enormously better than they are usually to-day doing.*

THE UTILIZABLE PART OF ENERGY STORED IN THE STEAM by

* Perkins employed a Woolf compound engine as early as 1827, and for his latest work with his highest pressures. Had Perkins and Alban, the distinguished German engineer who followed in his track, employed larger expansion ratios, and triple and quadruple expansion engines, it is more than possible that we should be to-day using, regularly and customarily, pressures exceeding these. Progress must apparently always be in that direction; we can see no way of increasing our engine efficiencies by reducing the lower limit of temperature and pressure.

transfer from the furnace, consisting, as it does, of the difference between the quantity of heat thus transferred and that rejected from the engine at the lower limit of temperature and pressure of its cycle, is measured, for the case of Carnot, by the ratio of the range of temperature worked through to the maximum absolute temperature, by $(T_1 - T_2) \div T_1$. In the common cycle here taken for comparison, this measure becomes the ratio of the difference of heats supplied and rejected to heat supplied, the ratio $(H_1 - H_2) \div H_1$; the two quantities being themselves measured from the total heat of the feed-water. The ratio of the two utilized quantities is approximately the ratio of total to latent heat from the same base point. With steam at Perkins's minimum pressure, 500 lbs. by gauge, and with feed at 140° Fahr. (60° C.), taking $J = 773$, the stored dynamic energy supplied by the furnace is, for example, 869,353 B. T. U.; and of such steam, 1,980,000 ÷ 869,353 = 2.28 lbs. per hour, contain the stored equivalent of one horse-power, at efficiency unity. In the Carnot cycle, the efficiency would be (470 − 140) ÷ 932 = 0.35; and the equivalent weight of steam would be 2.28 ÷ 0.35 = 6.4 lbs. per horse-power per hour. The heat transformed would be thus 7,174 B. T. U. per horse-power per hour, and at 9 lbs. evaporation, 0.7 of a pound of fuel per hour per horse-power. At 10 lbs. evaporation, this figure would be inside 0.65 lbs., about five-eighths of one pound.

For other cases it would be as follows:

MAXIMUM IDEAL EFFICIENCIES.

STEAM PRESSURES, ABSOLUTE.

250 Pounds per Square Inch.			100 Pounds per Square Inch.		
Efficiency.	Pounds per Horse-power per hour.	B. T. U. per Horse-power per hour.	Efficiency.	Pounds per Horse-power per hour.	B. T. U. per Horse-power per hour.
100.00	2.32	2,545	100.0	2.37	2,545
45.40	5	5,698	47.4	5	5,369
29.00	8	8,786	27.2	8	9,391
23.19	10	10,975	23.7	10	10,739
19.30	12	13,186	19.7	12	12,918
16.55	14	15,372	11.8	20	21,568
12.07	20	21,085	7.9	30	32,215

It will be noted that the weight of steam per horse-power and per hour is not the same for the same efficiencies, at different pressures, since, at each pressure, the heat stored per pound differs, becoming greater and greater for the common cycle, less and less for the Carnot cycle, and, for equal efficiencies, giving smaller weights for the former, as pressures rise, and greater for the latter. The number of thermal units, on the other hand, is constant for equal efficiencies at all pressures, 42.42 B. T. U., or 2545 B. T. U., per minute or per hour, divided by the efficiency.

THE THEORY OF THE CASE in hand may be taken to illustrate the ideal representative of a common good example of modern practice; the pressure being not at all uncommon with the contemporary engine, and close upon the margin between the usual range of the compound and that of the triple-expansion engine, above average practice for the one, below it for the other. With a pressure of 120 to 125 lbs. by gauge, 135 to 140 absolute, we find that, for efficiency unity, we shall require 2.3 lbs. of steam per horse-power per hour, and with such fuel as will give us 10,000 B. T. U. per pound, 0.2545 lbs. per hour per horse-power. The maximum thermo-dynamic Carnot efficiency would be 0.2714, and this would correspond to 8.66 lbs. of steam and 0.943 lbs. of coal per horse-power per hour.

Assuming the common—the Rankine—cycle, and computing efficiencies by the methods of Rankine, when expanding, with a ratio 10.62, down to 10 lbs. terminal pressure, and with 4 lbs. back-pressure, we obtain the results tabulated below. Beside them are placed the figures of similar import relating to the action of two other representative ideals, the one corresponding to that of performance of the old simple engine with large clearance, with a ratio of expansion of about 4; the other exhibiting the performance of the ideal Rankine cycle of the same initial pressure, but with a ratio of expansion fairly corresponding to what is usually regarded as an average best ratio for compound engines of the older designs, about 7. The comparison of these data with those to be obtained for the case taken as the subject of this paper, and their comparison with each other, will prove interesting.

IDEAL ENGINE EFFICIENCIES.

$p_1 = 135$; $p_2 = 4$; $T = 591°$ F., abs.; $D_1 = 0.306$; $r = 4.11, 6.93, 10.62$.

	Case I.	Case II.	Case III.
r	4.11	6.93	10.62
p_e	10080	6625	4970
Effic.	0.1613	0.1784	0.2048
B. T. U.	15,784	14,826	12,424
Steam	14.6	13.34	11.51
Fuel (St. ÷ 9)	1.6	1.48	1.28

To these computed consumptions of heat, steam, and fuel must be added, in the average case, perhaps from one-third to one-half for wastes of heat externally and internally, making the steam consumption, for the three cases, respectively, about 20, 18, and 15 lbs., and the fuel 2.2, and 2, and 1.7 lbs. per indicated horse-power per hour. The dynamometric, or brake, power being taken, as it properly should be, for the measure of useful transformation of energy, these figures would be increased about 15%, and would become, respectively, 23, and 20.7, and 17.25 lbs. of steam, and 2.6, and 2.3, and 1.92 lbs. of fuel per dynamometric horse-power per hour.

THE REPRESENTATIVE IDEAL CASE more exactly paralleling the real case to be studied, is the following: *

The thermo-dynamic Carnot efficiency, for this case, is, as a maximum for the perfect ideal engine, $(T_1 - T_2) \div T_1 = 0.283$; 28.3%. The case here studied is that of the jacketed engine, and it is assumed, with Rankine, that the jacket has precisely that effectiveness of action which insures the retention of the working charge in the dry and saturated state up to the point of rejection with the opening of the exhaust-valve. The formulas employed in these computations are those of Rankine, so far as applicable

* It may appear as interesting as it certainly is satisfactory to know that this case was computed, and the figures here given obtained, in advance of the trial with the results of which they are compared, and the estimated wastes were originally introduced as here given. The comparison of the estimates with the actual results of test are thus rendered somewhat striking. The computer was Mr. John R. Whittemore, M.M.E., at the time engaged in graduate work in Sibley College.

to the ideal case. The treatment of the wastes is that of the writer.*

The data assumed are the following, and are as nearly those of the trial as it was possible to make them, in advance.

DATA FOR COMPUTATIONS AND RESULTS.

$p_1 = 121.6 + 14.5 = 136.1$; $p_3 = 2.8$ lbs. per square inch : $T_1 = 811.8°$ F.

r	p_2	T_2	V_2	U	Therm. Efficiency.	Steam per H.P. per hour, W'.
15	7.68	642.2	48.93	206,600	0.232	9.61
20	5.55	628.0	66.40	214,600	.252	9.23
25	4.41	618.3	82.50	219,000	.258	9.02
30	3.63	610.5	98.90	220,380	.259	8.98
35	3.09	604.1	115.25	222,070	.261	8.92

THE WASTES for these cases may be approximately computed, and, if added to the above expenditures, the sum will presumably be approximately the actual expenditure of the engine, always provided the design and construction be so well and so intelligently carried out as to permit the operation of the machine to be representative of good contemporary practice. In the present case, there is not only no question of this fact; but we know that the performance of the real engine approximates more closely to the ideal than any case yet known. Our computations should, therefore, result in a larger, rather than a smaller, figure than is actually obtained for the expenditure of the engine during the trial.

The two formulas most commonly employed by the writer in the computation of probable wastes by cylinder condensation, the most remarkable of those wastes which distinguish the real from the ideal case, are the following. in which W is the weight of steam condensed per hour, and W' that computed for the ideal case:

$$W = CA\,(T_1 - T_2)\,t\,;\; W = (W'a\sqrt{rt}) \div D.$$

A and a are constants determined by experiment; t is the time of one stroke in seconds; and D is the diameter of the steam-cylinder in inches, A its internal superficies in feet.† As a

* For the Rankine equations, see either Rankine's *Steam-Engine and other Prime Movers*, or the writer's *Manual of the Steam-Engine*, Vol. I. For the treatment of the real, as distinguished from the ideal, case, see the latter, and especially Chaps. V. and VI.

† *Manual of the Steam-Engine*, Vol. I., Chap. VI.

matter of convenience, the second form was employed in checking results after test; but the first of these expressions was used in the antecedent computations; the temperatures being obtained on the assumption that the expansion was substantially the same in each cylinder. Taking the waste as being that of the cylinder most subject to this form of loss, and which our computations, as above, show to be, in this case, the low-pressure cylinder, we find, for $D = 74$; $r = 3$; computing for ideal steam and waste, respectively, $W' = 9.23$, as above, $W = 1.87$ lbs. of steam per *I.H.P.* per hour. The total loss by clearance spaces is 0.13 lbs.; that due to "drop" and intermediate losses of energy is 0.41, and the total is thus, neglecting external radiation, 11.64 lbs. of steam per indicated horse-power per hour. To this nearly 5% has been added by the computer for external wastes, and the total becomes 12.2 lbs. per *I.H.P.* per hour, thus:

Taking the mechanical efficiency of the engine as 90%, the expenditure of steam per dynamometric horse-power per hour, as computed, will be found to be 13.5 lbs.*

DISTRIBUTION OF ENERGIES.

Energy Supplied.	Per Cent.	Steam per Hour.	
		Per *I.H.P.*	Per *D.H.P.*
Useful work	23.0	2.80	3.10
Thermal wastes	74.8	9.13	10.12
Dynamic wastes	02.2	0.27	0.28
Total received	100	12.20	13.50

Conclusions.—One of the first points to be noted in connection with the performance of the boiler is its moderate efficiency, taken as a function of the ratio of area of heating surface to weight of coal burned in the unit of time. Computed by the Rankine formula,

$$E = BS \div (S + AF),$$

* As has been seen, the actual figures were, with boilers working with slightly moist steam, 11,678 lbs. per *I.H.P.*, and 12.86 lbs. per *D.H.P.*; the engine working with slightly less thermal efficiency than computed, but more than compensating this defect by an exceptionally low friction waste. In the contractor's trial these figures become 12.155 lbs. steam per *I.H.P.*, and 13.51 lbs. per *D.H.P.* With 2% external waste—a more reasonable figure than 5%—the figures are 11.87 and 13.2, and almost identical with the actual figures.

this quantity would be 90 to 94%, accordingly as one or another of his values of the coefficients is employed; the lower figure being that most accordant with the experience of the writer with anthracite coal, but the larger that given by the better qualities of bituminous coal, such as the fuel here used may be taken to represent. The figure actually obtained at the trial is seen to be, as given in the log (Appendix), but 73.45%, or but about 80% of the computed figure. This probable waste of approximately 20% of the fuel brings down its value as actually used to that of an anthracite containing over 20% of ash, and reduces its evaporation from a computed 11.2 lbs. to an actual 8.81, and from equivalent evaporation "from and at the boiling-point" of 12.8 down to 10.27; while it at the same time reduces the duty on actual coal consumption from what would have been above 160,000,000 to an actual 137,770,000. As already remarked, this difference is probably partly due to the fact that such slow combustion involves thin layers of fuel, and liability to dilution of the products of combustion by air; partly to the inefficiency of combustion of hydrocarbons. This loss, in this instance, more than balances the anticipated gain, by the wonderful freedom from ash and generally excellent character of the fuel chosen for the trial, and the exceptional dryness of the steam made.

The coal per square foot of grate, 7.32 lbs., is so low that, in the opinion of the writer and of many experienced engineers, but contrary, perhaps, to general opinion, the probability of excessive air-supply, as a consequence of the thinness of the fuel-bed, and with resulting reduction of furnace efficiency, gives reason to think that better work might have been done with the grates bricked off somewhat, or even with a smaller number of boilers. The steam made per square foot of heating surface was but 1.49 lbs., or, "from and at 212," 1.73; the factor of evaporation being 1.154. The quality of steam was high—98.95—a point which is very carefully provided for by the designer, and probably makes up, in a very sensible degree, for the low evaporation rate; both having a common origin in the slow combustion of the fuel.

The dry steam, the high pressure, the large expansion ratio, the low condenser-pressure and temperature, the inappreciable air-leak into the condenser, the excellent valve-gear and well-proportioned ports, the very unusually small clearances, combine to give us here the closest approximation to the ideal case that has yet been recorded. The indicator-diagrams, which have been else-

where reproduced, correspond with extraordinary closeness to the ideal; and the loss of work by discrepancy between the ideal and the real diagram is as remarkably small. (Fig. 106.)

The mean effective pressure, for the engine, reduced to low-pressure cylinder volume, is 22 lbs.; each pound gained by improved vacuum is thus a gain of $1 \div 22 = 0.0455$, or 4.55% in the work of the engine; of which a small percentage, again, must be returned in the operation of the air-pump. Four per cent. per pound of pressure, thus gained, is considered by the designers of the engine well worth all the care, skill, and conscientious supervision of the operation of the engine required to obtain and to maintain it. The slight compression observed is made less effective by the low terminal pressure; but it has a sensible effect in reducing the slight loss by clearance.

The low friction loss of the engine is due to a number of concurrent circumstances.* The cylinders are vertical; they are coupled, piston to piston, with the pumps; the wheel and shaft have only the friction of their own weight and of the small, varying work of regulation; the triple-crank arrangement makes regulation easy and shaft friction a minimum; the circulating pump works in a solid column, demanding very little power in its movement; the stuffing-boxes are deep, and thus need be but lightly packed; and the whole machine is designed, constructed, and operated with evident regard to the relative magnitude and importance of the partially avoidable wastes, both dynamic and thermal, of the whole machine. Perhaps one of the most interesting of the minor points noted in the whole investigation is the gradual reduction with time of the dynamic waste; the friction falling from about 14% on starting the new engine and at the contractor's, the Benzenberg and Lewis, trial, to 9% at the Sibley College trial, eight months later.

THE DISTRIBUTION OF ENERGY supplied the engine, during the trial, can now be readily determined by comparison of the real with the ideal case. It is seen by reverting to the estimates already given, that the ideal engine, under similar circumstances,

* The friction of the engine is remarkably low for this type of pumping engine; but it is, of course, still above the figure obtained from the best direct-acting machines, which, in the Newton, Mass., trial, reported recently, for example, was found to be 4.2%, giving a mechanical efficiency of 0.958, as compared with that here obtained of 0.9078. But the latter figure excels, on the other hand, probably, that obtained from the average direct-acting machine.

10

14.5

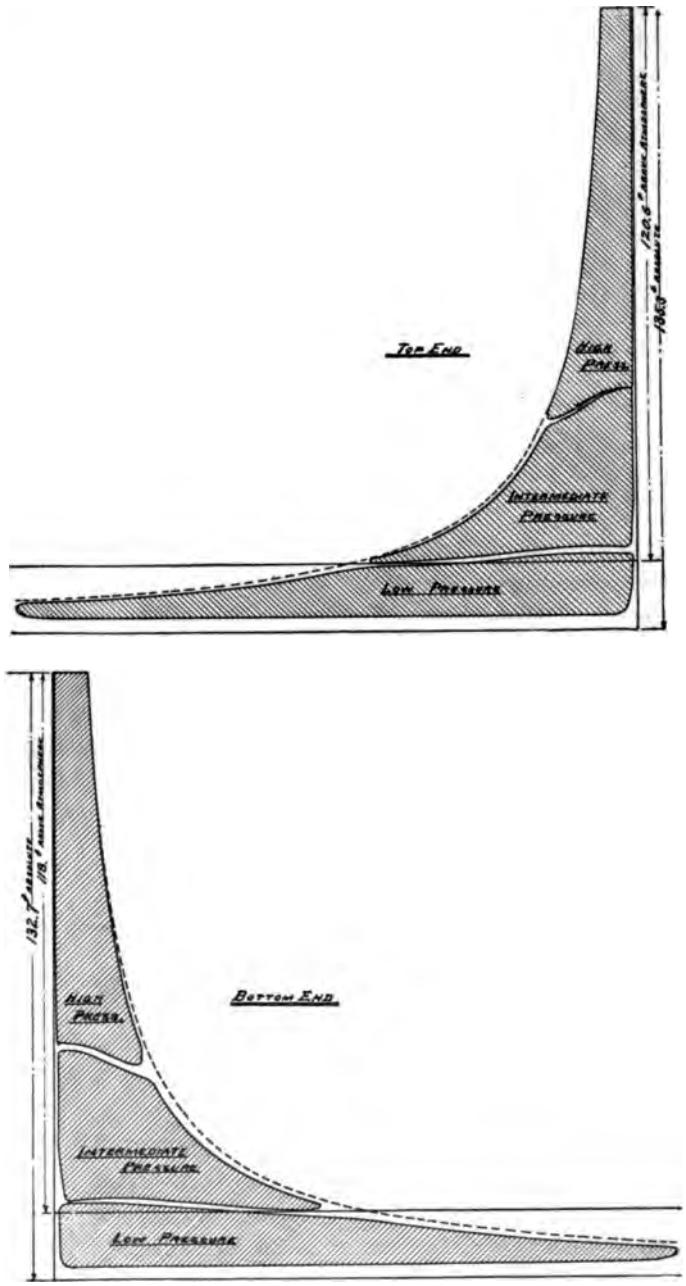


FIG. 106.—COMBINED INDICATOR DIAGRAMS

would have an efficiency of about 0.25, and would require 9.23 lbs. of steam, nearly a pound of fuel per horse-power per hour. We find, from the records of the trial, that of all the energy developed in the cylinders, 9.22% was wasted in friction. We thus obtain the following figures to compare with those computed, page 360:

DISTRIBUTION OF ENERGIES.

Part of Total Energy supplied.	Per cent.	Per D.H.P., per Hour.	
		B.T.U.	lbs. st.
Thermal and thermo-dynamic wastes...	80.57	11688.25	10.455
Dynamic, or friction, waste.....	1.80	258.87	0.232
Useful work	17.63	2535.55	2.267
Total thermo-dynamic transformation..	19.43	2794.42	2.499
Total from boilers..... ...	100.00	14,882	12.864

Thermo-dynamic Condensation, or "adiabatic condensation," as it is often called, produces a continued fall in pressure, which reduces the power which would otherwise be given by the engine, although it is compensated by the fact of the transformation of this stored heat-energy. The amount of this condensation is quite considerable in cases of large expansion ratios like the present. Computing this amount by the familiar expression constructed by Rankine, we obtain the results illustrated in the accompanying diagram (Fig. 107); the larger values being obtained by the use of the exact formula,

$$(v - u) \div v = 1 - \frac{T}{H}\left(J \log \frac{T_1}{T} + \frac{H_1}{T_1}\right),$$

and the smaller values come from the use of the approximate expression given by the same author for the adiabatic expansion-curve, comparing its abscissas with the volumes of dry and saturated steam as given in the "steam tables." It is found that, except for the action of the jackets, which, in the Milwaukee engine, cover heads and sides of all the cylinders, the thermo-dynamic condensation alone would convert about 20% of the entering steam into water. The curves here drawn show the progressive liquefaction of the steam, as the pressure is worked down from 140 lbs., absolute, to lower tensions.

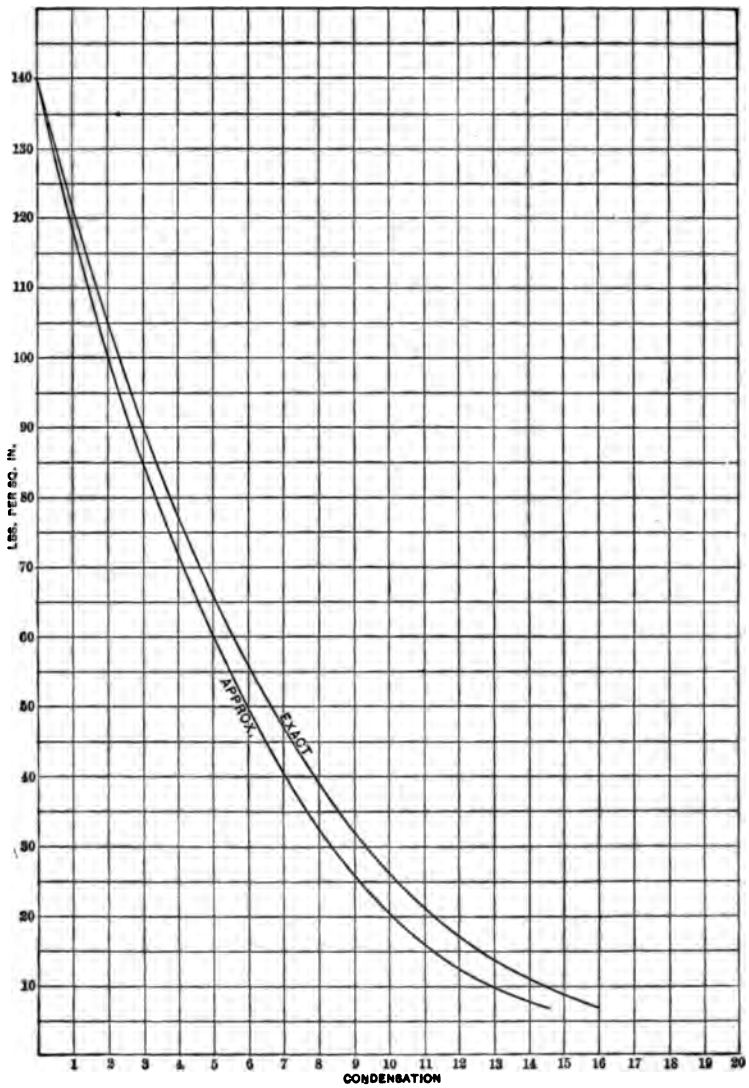


FIG. 107.—THERMO-DYNAMIC CONDENSATION OF STEAM.

PROGRESSIVE THERMO-DYNAMIC CONDENSATION.

Pressure lbs. per sq. inch from $p = 0$.	Temp. Fahr. absolute, from $T = 0$.	FRACTION OF TOTAL STEAM CONDENSED, THERMO-DYNAMICALLY.	
		Approximate.	Exact.
140	814°	0.000	0.000
130	808	.005	0.0055
115	799	.012	.014
100	790	.019	.022
90	781	.026	.030
80	773	.034	.038
70	764	.041	.046
60	754	.049	.055
55	746	.055	.062
45	735	.065	.070
40	728	.070	.078
35	720	.078	.086
30	710	.086	.094
25	699	.093	.100
20	692	.100	.111
18	682	.108	.118
15	673	.115	.126
12	664	.123	.136
10	655	.131	.144
8	644	.137	.155
7	638	.145	.159
6	629	.150	.165
5	620	.154	.169

THE FUNDAMENTAL PRINCIPLES OF THE MULTIPLE-EXPANSION ENGINE are closely observed in the design of this machine. The low cylinder-condensation shows not only that the jackets were well-designed and efficient, but that the wastes of the high-pressure and intermediate cylinders do not flood the low-pressure cylinder and produce unnecessary loss. The distribution of power among the several cylinders is excellent; the high-pressure and intermediate having substantially equal powers, and the low-pressure owing its higher power to the addition of the full-stroke accession of effort due to its communication with the condenser. The exceptional efficiency of the machine may probably be taken as proof that it is, in cases of this kind, which are of exceptionally good design and construction, economically advisable to make the number of cylinders chosen a maximum for the pressure adopted.* The total ratio of expansion was very nearly 19 in this case, and the ratio for each, assuming equality, was thus $\sqrt[3]{19.205} = 2.7$, the apparent cut-off being variable between 0.32

* *Manual of the Steam-Engine*, Vol. I., p. 597, *et seq.*

and 0.39. The terminal pressure is also here exceptionally low; the usual best figure being considered to be, with good engines, not below 6 lbs. absolute; it is more commonly 8, and often 10. Here it is made as low as 5.3, and the back-pressure is but 1.63 lbs., or 0.11 atmosphere; the drop between cylinder and condenser being unusually slight.

JACKET ACTION is here very effective. The measurements of the indicator diagrams, as originally practiced by Hirn, show a progressive increase, in the internal condensation, of from 12% in the first and 14% in the intermediate cylinder, to 18% in the low-pressure cylinder. The difference between the initial and the final condensation is $18 - 12 = 8\%$, or about one-half the thermo-dynamic condensation; this indicating that "cylinder condensation," finally, amounted to about 5%, in spite of the effectiveness of the jackets. If we take the waste from external distribution of heat to be 2%, and the condensation in the jacket to be due to internal absorption of heat to the extent of 7.25%, it would appear that the heat expended in the jackets was at least twice as effective—perhaps in much larger proportion, the exact amount is indeterminable—as that worked in the steam cylinders. The observed condensation increases 2%, cylinder by cylinder; the normal thermo-dynamic, or adiabatic, condensation would be 5%, similarly cumulative; and the actual initial cylinder condensation is found to be 12%. It thus appears that the jackets reduced the latter quantity, which, in an unjacketed engine of this size and action, would, as known by experience, be at least 25%, and possibly 30, to 12% initially, and to a final—which would have been 30 or 40—of 20. This heat-transfer from the jackets is equivalent to a superheating of about 100° Fahr. They thus saved, net, not less than 10%. This, taken as a fraction of the $16,410 reported by the officers of the city water department as the annual cost, amounts to $1,641, the interest on $32,820, and on probably ten times the actual increased cost of engines due to jackets; otherwise stated, it is probably 50% interest on the investment in jackets.

The quality of steam supplied by the boiler being taken at 0.99 at the engine, 0.87, 0.85, and 0.81, respectively, in the several cylinders taken in order, the initial condensation is seen to have been 12% in the first, 14% in the second, and 18% in the third cylinder, thus showing, as seems to have been almost unquestionably the fact, that the jackets were able to more than compensate the progressive deterioration of quality in the engine

due to thermo-dynamic condensation of steam. The jackets therefore reduced the cylinder condensation due to the action of the sides of the cylinders from what would probably have been 30%, perhaps 40%, of the total supplied, by holding the quality of the working steam up to approximately 0.80 at entrance into each cylinder.

The tables appended, containing the results of the application of Hirn's Analysis to this case, and representing the first complete analysis yet made of a commercial machine of this character, have double interest in the fact that they exhibit so fully the conditions affecting the flow of energy through the engine, and its distribution into useful and wasted work, and at the same time are representative of this distribution as affected by the best of contemporary designing, and as attaching to the most successful instance on record, to date, of thermo-dynamic transformation in commercial, or other, applications.*

In making these computations, the fact that the low-pressure cylinder received an excess of heat from the jackets led to some embarrassment in the preliminary work; but the analysis is so complete that no difficulty was experienced in tracing out the source of the derangement of the usual process of analysis, and the tables exhibit the whole course of the received, the distributed, and the applied and rejected energies, in each cylinder, and from cylinder to cylinder, up to their final discharge, into the condenser and the air, so far as rejected.

The loss by external conduction and radiation was determined by the expression,

$$R = 1 \div 3\ [Q + J - (K + K') - \Sigma\ W]\ ;$$

in which

Q = total heat supplied;
J = heat given up by jackets;
$K + K'$ = heat rejected from engine;
$\Sigma\ W$ = work developed in the engine;
R = radiation-loss from each cylinder.

* The computations of which the data and results are here published were made for the writer, in this instance, by Messrs. S. H. Barraclough (B. E., Univ. Sydney, N. S. W.) and L. S. Marks (B. S., Mason Coll. and Lond. Univ., G. B.), graduate students engaged in two years' advanced work and research, preparatory to taking the higher degree in engineering in Sibley College, Cornell University.

The heat rejected from the first and second cylinders was in each case taken as the heat received, as obtained from the indicator-diagram, less the work performed plus the radiation. That rejected from the low-pressure cylinder was obtained from a special supplementary trial, as already noted.

The main points of interest, in addition to those already observed, and connoted with the latter, are the following :

Steam entered the engine nearly dry, having a quality 0.9895, dry steam being taken as unity ; 5.56 lbs. being expended per revolution, in addition to the .0462 lbs. encaged in the clearance-space, which latter, however, was extraordinarily small, 1.4%. At 20.3133 revolutions per minute, this developed 573.87 I.H.P. on an expenditure of 11.678 lbs. of dry steam per hour per horse-power, probably the lowest yet recorded.

Jacket action may, perhaps, be more clearly observed in the results attained in transfers of heat to cylinders from the jackets, as seen in the Hirn analysis given in Appendix D, which traces the progress of only that portion of the expended steam which actually enters the cylinders from the boiler, and exclusively of the steam used in the jackets. The figures given in Table C cover the whole steam supply, and, on that account, exhibit this difference less perfectly. The steam used per 100 strokes is taken, as before, and we find that of the 556.4 lbs. sent to the engine from the boiler, 504.5 actually enter the cylinders and pass through them to the condenser ; receiving heat, meantime, from the 52 lbs. of jacket steam, 9¼% of the total steam supplied, which is condensed in the jackets to supply that heat. The engine receives 598,800 B. T. U. with the steam entering it, and 40,870 from its jackets, a total of 639,670 B. T. U. Of this, 119,652 B. T. U. is transformed into work and the remainder wasted. Of this remainder, 6,598 units are radiated from the exterior of the engine. The jackets supply 13,350 to 14,150 B. T. U. to each cylinder, the quantity increasing, as would be anticipated, with fall of temperatures and pressures.

The quality of the entering steam is 98.95%, which becomes 86.77 at cut-off by the initial condensation of 12.18% by the chilling action of the cylinder walls. Reëvaporation and the heat from the jacket conspire to raise this figure to 93.3 at release, and to 96.8 during exhaust from the high-pressure cylinder. At cut-off in the intermediate cylinder, this drops to 85.4 again ; the jackets compensating the thermo-dynamic, adiabatic condensation very nearly, and the defect of quality, as compared with the

steam in the high-pressure cylinder, being produced substantially by external wastes. At release, the quality becomes the same as at the same point in the first cylinder; the jackets having, by this time, compensated the external wastes as well. In the low-pressure cylinder the quality 94 at entrance becomes 81 at cut-off, external wastes having apparently somewhat larger influence than previously, and thermo-dynamic condensation showing very much more of its cumulative effect. The steam sent to the condenser, finally, has a quality of 89.95%, showing clearly the result of the opposing actions of jacket heat and thermo-dynamic and thermal wastes and condensations. The negative values, in each table, show that each cylinder, on the whole, gains more than it loses—those quantities representing heat transferred to the working charge—and the magnitudes of the quantities of heat entering from the jackets during expansion and exhaust are seen to be amply sufficient to account for the jacket effect observed in the final efficiency.

In the tables, line 9 gives the quantity of heat supplied the engine in the steam entering the working cylinder, and the three sheets show its gradual variation as it passes through, losing energy in thermal form by external wastes and in dynamic form by conversion into work. In each case, as seen on lines 23 to 29, more heat enters than leaves the cylinder, the gain being due jacket action and amounting to about 2%, net. The reëvaporations in the cycle, subsequent to cut-off, and accessions of heat from the jackets, lines 19, 20, are greater, as a total, during expansion than during exhaust in the high-pressure cylinder, while this relation is reversed in the intermediate and large cylinders, showing the greater probable advantage of jacketing the former in cases similar to that here studied. This transfer during exhaust is wholly a waste; it is in part only a gain during expansion. The jacket is evidently only valuable by its effective restriction of initial condensation. Its action is wholly unsatisfactory during the entire cycle otherwise, for even the heat added by it during expansion is thermo-dynamically undesirable. It would be better if it were received direct from the boiler and at the maximum temperature of the prime steam. The fact of an excess of heat being rejected from the cylinder is only satisfactory as an evidence that the cylinder walls are dry, and presumably, also, warmed by the jacket, which latter fact is shown by the extent to which thermal interior waste is reduced in this engine. The exceptionally low total of wastes, corresponding to two and a half pounds of steam, or less than 30% of the computed demand

of the ideal engine, and which represents the remaining margin for gain in future constructions, is largely due to this effective phase of jacket action.

It is to be noted that the reduced pressure in the low-pressure jacket makes the relative heat supply vary somewhat peculiarly. The high and intermediate cylinders were jacketed, like the receivers, with boiler steam. In the low-pressure jacket the pressure was reduced to 34 lbs. by gauge.

Were the engine structurally perfect, it would have demanded 9.23 lbs. of steam per H. P. per hour. The additional 2.45 lbs. per I. H. P., 3.63 lbs. per D. H. P., measure the still remaining range for improvement. The engine, as a whole, has two-thirds of the efficiency of the ideal thermo-dynamic machine. The thermo-dynamic efficiency of the machine, as a whole, was 27.14%, and its actual efficiency 19.4%, all thermal losses included. This is a wonderful result, and the designer, the builders, and the proprietors of this extraordinary engine are entitled to congratulation, not only for such great success, but for having thus shown what can be done by good designing, good construction, and good management, by the most intimate union of scientific and practical wisdom and art, toward the perfection of the steam-engine, in the approximation of the real to the ideal.

APPENDIX A.

NORTH POINT PUMPING STATION; 1875.

Test made by W. E. Worthen, Thos. Whitman, and Chas. Hermany.

Engine No. 1 and 2.—May 19th to 21st, 48 hours. Duty, 76,955,720 ft. lbs.
Under same conditions, 1884, 24 hours. Duty, 77,049,906 ft.-lbs.
With Receiver cutting off in Low, 24 hours. Duty, 85,398,586 ft.-lbs.
Engine No. 3.—July 6th to 8th, 1882, 48 hours. Duty, 104,820,341 ft.-lbs.
Engine No. 4.—July 12th and 13th, 1892, 24 hours. Duty, 126,865,240 ft.-lbs.

Year.		Gallons.
1891.	Pumped, No. 1 and 2	4,320,989,960
1891.	Pumped, No. 3	3,021,633,442
1891.	Pumped, No. 4	1,414,381,250
	Total	8,756,954,720

Average Duty from total coal consumed, 80,428,844 ft.-lbs. Coal consumed for pumping, 13,875,610 lbs. Coal, total consumed for all purposes, 14,113,410 lbs. Average Duty, coal consumed for pumping, 81,807,376 ft.-lbs.

Year.		Gallons.
1892.	Pumped, No. 1 and 2	909,080,380
1892.	Pumped, No. 3	2,192,715,134
1892.	Pumped, No. 4	5,204,368,750
	Total	8,306,164,264

Total amount of coal used for pumping, 11,113,200 lbs. Duty from above coal, 99,043,048 ft.-lbs.

The above coal includes heating and lighting building, running shop engine, and pumping out wells.

(Signed) THOMAS MCMILLAN,
Engineer.

APPENDIX B.

Dimensions, Logs, Data, and Results of Trial of the Milwaukee Pumping Engine.

Dimension of Boilers.

Length	18 feet.
Diameter	66 ins.
Grate	5 ft. long, 5 ft. 4 ins. wide.
Grate surface	26.65 sq. ft.
Tubes in each boiler, 55	diameter 4 ins.
Area of tubes in each boiler	4.68 sq. ft.
Water-heating surface	1,216 sq. ft.
Ratio of heating to grate surface	35
Height of chimney	125 ft.
Area of chimney	12.566 sq. ft.
Total grate surface of the four boilers	106.6 sq. ft.
Total heating surface	4,864.0 sq. ft.
Bottom of gauge glass to bottom of boiler shell	3 ft. 1¼ ins.

Boiler Test.

Duration of test	24 hours.
Barometer inches of mercury	29.54
Atmospheric pressure, pounds per sq. in	14.5
Steam gauge, corrected (at engine)	121.6
Draft gauge, inches of water	0.4025
Absolute steam pressure	136.1 lbs.
External air, temperature deg. F	25.71
Boiler room, " "	50.
Flue, Thermometer in top, deg. F	406.02
Flue, Pyrometer in center, "	403.02
Feed water	97
Steam	350.6

Fuel.

Total coal consumed, pounds	18,234
Moisture in coal, per cent	5.25
Dry coal consumed	17,277
Total ash, dry, pounds	255
Total ash, dry, per cent. of dry coal	1.47
Moisture and ash, per cent. of wet coal	6.65
Total combustible, pounds	17.022

Fuel per Hour, Pounds.

Actual	759.75
Dry coal	719.84
Combustible	709.25
Actual coal per sq. ft. of grate	7.15
Dry coal " " "	6.76
Combustible " " "	6.65

Calorimeter.

Temperature of steam in calorimeter, deg. F	284.6
Back-pressure in calorimeter, inches of mercury	1.9
Quality of steam	98.95
Per cent. of entrained water	1.05
Number of minutes calorimeter was open	114
Steam used in calorimeter, pounds	208.5

Feed-Water.

Total weight of water used, pounds	162,864
Excess of water in boiler at end of run, pounds	131
Total evaporated dry steam, pounds	161,023.2
Factor of evaporation. (Equivalent value of 1 lb. wet steam to 1 lb. dry steam from and at 212 F.)	1.154
Total equivalent evaporation from and at 212° F., pounds	187,794

Feed-Water per Hour, Pounds.

Actual amount used, pounds	6,780.5
Evaporated dry steam	6,709.4
Equivalent evaporated from and at 212°	7,710

Evaporation per Pound of Coal, Pounds.

* Apparent, feed-water 97°, steam 121.4	8.906
† Actual, to dry steam	8.81
Equivalent from and at 212°	10.27

Evaporation per Pound of Dry Coal, Pounds.

* Apparent, feed-water 97°, steam 121.4	9.425
Equivalent from and at 212°	10.72

Evaporation per Pound of Combustible, Pounds.

* Apparent, feed-water 97°, steam 121.4	9.56
Equivalent from and at 212°	10.88

Evaporation per Hour.—Per Square Foot of Grate.

Actual, uncorrected for moisture, pounds	63.6
Equivalent from and at 212°, pounds	73.2

Per Square Foot of Water-Heating Surface.

Actual, uncorrected for moisture, pounds	1.505
Equivalent from and at 212°, pounds	1.73

* Uncorrected for calorimeter. † Corrected for calorimeter.

Per Square Foot of Least Draught Area.

Actual, uncorrected for moisture, pounds	373
Equivalent from and at 212°, pounds	429

Horse-Power.

On basis 30 lbs. from 100° F. to 70 lbs. pressure	223.5
Builder's rating	400
Ratio of boiler horse-power to capacity, per cent	55.7

Efficiency of Boiler.

(*A*) Heat generating per hour on basis of 14,500 B. T. U. each pound of combustible	10,285,125
(*B*) Heat absorbed by steam per hour, B. T. U	7,554,838
Efficiency of boiler, (*A*)÷(*B*), per cent	73.45

TRIPLE-EXPANSION PUMPING ENGINE.

Dimensions.

Length of stroke of each piston	60 ins.
Diameter of high-pressure cylinder	28 "
Diameter of intermediate-pressure cylinder	48 "
Diameter of low-pressure cylinder	74 "
Diameter of piston-rod in each cylinder	4 "
Number of piston-rods " " "	6 "
Clearance high-pressure cylinder, per cent	1.4
" intermediate-pressure cylinder, per cent	1.5
" low-pressure cylinder, per cent	0.77
Volume of first receiver	101.3 cu. ft.
" second receiver	181 cu. ft.
Number of reheater pipes, first receiver	57
" " " second receiver	35
Diameter reheater pipes	2 ins.
Number of single-acting water plungers	3
Diameter of each	32 ins.
Diameter of single-acting air-pump	20 "
Diameter of single-acting plunger feed pump	2¼ "
Diameter of single-acting air compressing pump	2¼ "
Diameter of double-acting circulating pump	7½ "
Stroke of all pump plungers	60 "
Distance from center of pressure gauge to center of pump chamber	19.8 ft.
Distance from bottom of well to center of pump chamber	20 ft.

Area of Piston, Square Inches.

	Top.	Bottom.
Area of piston, high pressure	615.745	590.621
" " intermediate pressure	1,809.562	1,784.429
" " low pressure	4,300.85	4,275.72
Area of each pump plunger	804.2496 sq. in.	5.585 sq. ft.
Total volume delivered per revolution by one plunger		27.9253 cu. ft.
" " " " " three "		626.688 gals.
" weight " " " " "		5,229.291 lbs.

ENGINE TEST.—Results and Data.

Duration of test	24 hours.

Average Temperatures.

Water at pump well, F	34°
Feed water to boiler	97°
Discharge from air-pump	105.4
Calorimeter (1.9 inches back-pressure)	284.5
Engine room	69.9
External air	25.71

Average Pressures.

Barometer, inches	29.54
" pounds	14.5
Gauge at throttle	121.45
Absolute pressure at engine	135.94
Vacuum gauge, pounds	13.84
First receiver gauge, pounds	32.43
Second receiver gauge, pounds	1.3
High-pressure jacket, pounds	121.40
Low pressure by gauge, pounds	56.47
Suction head by float, feet	10.77

Revolutions.

Total number	29,252
Per hour	1,218.8
Per minute	20.314
Quality of steam, per cent	98.95
Moisture in steam, per cent	1.05

Feed-Water and Dry Steam.

Total feed-water to boiler, pounds	162,864
Excess in boilers at end of run, pounds	131
Steam used by calorimeter (114 min.), pounds	208.5
Total wet steam to engine	162,524.5
" dry " "	160,818.1
Wet steam to engine per hour	6,771.8
Dry steam to engine per hour	6,700.7
Heat in one pound wet steam above 105.4 F., B. T. U	1,106.26
Heat supplied engine per hour, B. T. U	7,493,444
Heat supplied engine per minute, B. T. U	124,890.7
Total wet steam used in jackets	15,054
Wet steam used in jackets per hour	627.3
Per cent. of total steam used by jackets	9.25

Indicated Horse Power.

		Uncor'ctd.	Cor.
High-pressure cylinder	Top	89.698	87.54
	Bottom	89.802	87.85
		178.498	175.39
Intermediate	Top	85.75	88.84
	Bottom	78.5	80.78
		164.25	169.62
Low	Top	118.66	116.23
	Bottom	107.36	112.63
		226.02	228.86
Total		568.668	573.87

Delivered horse-power	520.96
Friction horse-power	52.91
Friction horse-power per cent	9.22
Dry steam per I. H. P. per hour	11.678
B. T. U. per I. H. P. per minute	217.6

Relative Volumes of Cylinders.

High pressure	1	1
Intermediate	2.95	3.01
Low	6.95	7.01
Total number of expansions	19.55	

Water per I. H. P. per Hour from Diagrams.

Computed on supposition that all work is done in L. P. cylinder.

	High.	Intermediate.	Low.
Equivalent cut-off, per cent. of stroke in L. P. cylinder	4.85	14.05	39.07
Equivalent release, per cent. of stroke in L. P. cylinder	14.00	42.05	98.0
Equivalent clearance in L. P. cylinder	0.2	0.65	0.77
Steam per I. H. P. per hour, from diagram, at cut-off	9.35	9.12	8.37
Steam per I. H. P. per hour, from diagram, at release	10.1	10.0	8.92
Steam accounted for by indicator at cut-off, per cent	87.1	85 0	78.2
Steam accounted for by indicator at release, per cent	94.0	93.2	88.2

PUMP TEST.

Duration	24 hours.
Temperature of water pumped, deg. F	34
Weight per cubic foot	62.42
Pounds of water lifted per revolution	5,229.2917
Water pressure in pounds	56.473
Delivery head in feet	131.275
Distance delivery gauge to center of pump feet	19.8
Suction head in feet	10.77
Total head in feet	161.845
Revolutions per hour	1,218.8
Foot-pounds of work per hour	1,031,520,000
Kilogrammeters per hour	142,556,000
Wet coal per hour, pounds	759.75
Dry coal per hour, pounds	719.8
Kilograms of coal per hour	332.23
Combustible per hour, pounds	709.25

Capacity.

Cubic feet per revolution 3 × 27.9253	83.7759
Gallons per revolution	626.688
Cubic feet per hour	102,122.5
Gallons per hour	763,807
Cubic feet for 24 hours	2,450,940
Gallons for 24 hours	18,331,364

PRINCIPAL QUANTITIES OF BOILER AND PUMP TEST FOR EACH WATCH OF FIREMAN.

Number of Watch.	A.	B.	C.	A.	Total or Averages.
Duration of hours	5	8	8	3	24
Coal fired	3591	6284	6284	2075	18234
Coal per hour fired	718	785.5	785.5	691.60	759.75
Estimated excess in furnace	−35	110	150	45	.....
Coal per hour burned	725	775	776	730	.....
Dry coal (cor. for 5.25 per ct.)	688	735	736	693	719.8
Temperature feed water	.....	.....	.....	.....	.....
Steam pressure at engine	121.43	122.14	121.29	120.41	121.46
Barometer, inches	29.55	29.53	29.52	29.577	29.537
No. bbls. water per watch:					
Bbl. No. 1, net weight 400¼	45	60	63	24(-20lb)	192−20 lb
Bbl. No. 2, net weight 386¾	45	60	62	24	191
Bbl. No. 3, net weight 173¾	13⅛	24	23	10	70⅛
Weight of water, Bbl. No. 1	18011	24016	25216	9586	.....
Bbl. No. 2	17016	23595	23978	9282	.....
Bbl. No. 3	2874	4164	3591	1735	.....
Total feed-water	37301	51775	53185	20603	162864
Excess in tank, end watch	735	750	300	.....	.....
Excess in boilers (change in ht. on gauge glass, in.)	1.75	0.16	−0.215	0.08	.....
Excess in boilers, lbs	2667	259	−365	131	131
Total evaporated during watch	33849	54218	54357	20517	54354
Evaporated per hour	6770	6777	6795	6805	6780.5
Evap. per lb. of coal, actual	9.36	8.74	8.74	9.35	8.906
Calorimeter open, minutes	45	44	.....	25	114
Steam used by cal., lbs	82	81	.....	45.5	208.5
Quality of steam per cent	98.95	98.95	98.95	98.95	98.95
Temp. of water pumped	34	34	34	34	34
Water pressure in lbs	56.775	56.86	56.96	57.08	56.903
Deliv. hd. in ft. (2.307 ft. = 1 lb.)	130.96	131.25	131.31	131.56	131.275
Distance center pump to gauge	19.8	19.8	19.8	19.8	19.8
Suction head in feet	10.7	10.594	10.398	11.005	10.77
Total head in feet	161.46	161.644	162.008	162.365	161.845
Millions of ft.-lbs. per hour	1025.008	1032.6	1032.723	1034.995	1031.520
Actual duty from 100 lbs. dry coal burned, millions ft.-lbs	150.06	140.08	140.02	150.07	143.30647
Actual duty from 100 lbs. coal burned, millions ft.-lbs	142.7106	133.238	133.0827	142.7	135.77
Duty from 1,000 lbs. of steam, millions ft.-lbs	152.45	152.4	152.3	152.5	152.448

Note.—Fireman "A" was on five hours at beginning and three hours at end of test. "B" and "C" each eight hours. "A" carried thin fires; "B" and "C" thick fires. The condition of fires at end of each watch was estimated as follows, compared with conditions at starting:

First watch, (− 35 lbs.); second watch, 110 lbs.; third watch, 150 lbs.; fourth watch, 45 lbs.

These results do not have a probable error exceeding one-half of 1%. The last column gives the total average for the whole run, from which no corrections for excess of fuel are deducted. During the entire test draught was regulated by upper damper.

APPENDIX C.

APPLICATION OF HIRN'S ANALYSIS.

[FORM No. 1.]

SIBLEY COLLEGE, CORNELL UNIVERSITY.

DATA AND RESULTS.

Test of steam engine made by E. P. Allis Co., at Milwaukee, Wis.

Which engine—high pressure cylinder.
Diameter of cylinder, 28 inches.
Length of stroke, 60 inches.
Diameter of piston rods (2), 4 inches.
Volume of cylinder, crank end, 20.53.
Volume, head end, 21.40.
Volume of clearance, cu. ft., head, .298.
Clearance in per cent. of stroke, 1.4.
Volume of clearance, cu. ft., crank, .287.
Clearance in per cent. of stroke, 1.4.
Boiler pressure by gauge, 121.4.
Barometer, 14.5.
Boiler pressure, absolute, 135.9.
Revolutions per hour, 1,218.8.
Quality of steam in steam pipe, 98.95.
Weight of condensed steam per hour, 6,771.8.
Steam used during run, pounds, 162,-524.5.
Lbs. of wet steam per revolution, 5.56.
Per stroke, head, 2.72.
Per stroke, crank, 2.84.

SYMBOLS.

To denote different portions of the stroke, the following subscripts are used: Admission (a); expansion (b); exhaust (c); compression (d).

To denote different events of the stroke, the following sub-numbers are used: Cut off (1); release (2); compression, beginning of (3); admission, beginning of (0); in exhaust (5).

Quality of steam denoted by x.

Cut off, crank end per cent. of stroke, 34.1.
Cut off, head end per cent. of stroke, 33.4.
Compression, crank end per cent. of stroke, 1.73.
Compression, head end per cent. of stroke, 2.89.
Release, crank end, 100.
Release, head end, 100.
Pounds of dry steam (3 cylinders) per total I. H. P., 11.678.
I. H. P., head, 87.54.
I. H. P., crank, 87.85.
Total I. H. P., 175.39.

HIGH PRESSURE.

[FORM No. 2.]

ABSOLUTE PRESSURES FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

		Cut-Off.	Release.	Beginning.		Symbols. Rankine.	Symbols. Clausius.
				Compression.	Of admission.		
Subscripts used		1	2	3	0		
Absolute pressure	Head	134.2	45.0	48.2	135.3		
	Crank	132.4	48.7	49.7	132.7	P	p
Heat of liquid	Head	320.63	243.68	248.11	321.27		
	Crank	319.56	248.66	249.97	319.74	S	q
Internal latent heat	Head	786.01	844.80	841.46	785.52		
	Crank	786.84	840.98	840.12	786.69	I	ρ
Latent heat evaporation	Head	867.92	921.93	917.90	867.46		
	Crank	868.67	918.44	916.50	868.55	L	r
Total heat	Head	1,188.55	1,165.61	1,166.00	1,188.73		
	Crank	1,188.23	1,167.10	1,167.47	1,188.29	H	λ
Volume 1 lb. cubic feet	Head	3.290	9.207	8.684	3.265		
	Crank	3.332	8.550	8.350	3.295	C	v
Volumes symbols		$V_c + V_1$	$V_e + V_2$	$V_c + V_3$	$V_0 + V_e$	V_e	..
Volumes head, cubic feet		7.579	21.68	.668	.299	.299	..
Volumes crank, cubic feet		7.434	21.68	.916	.299	.299	..

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End.		Crank End.	
		Mean Pressures.	External Work. B. T. U.	Mean Pressures.	External Work. B. T. U.
Symbols		MEP	AW*	MEP	AW*
Admission	a	118.9	157.1	125.4	162.5
Expansion	b	58.6	155.7	62.8	157.2
Exhaust	c	32.68	129.0	34.81	129.7
Compression	d	24.15	2.765	43.7	6.48
Total, net		.....	181.035	.....	183.52

$A = \frac{1}{778}$. V_e = volume in clearance spaces.

HIGH PRESSURE.

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 STROKES.

JACKET STEAM INCLUDED.

Quantities.	Symbols.	Formulæ.	Head.	Crank.
Steam from boiler, pounds	M		272	284
Steam in clearance, pounds	M_c	$100\,(V_c+C_cX_c)$	9.16	8.998
Steam at admission, pounds	M_0	$100\,\frac{V_c+V_0}{C_0X_0}$	9.16	8.998
Steam, total, pounds	$M+M_0$		281.16	292.998
Heat of condensed steam	K'	MS_g		
Condensing water, pounds	G	$K+K^1=$	287,061	303,761
Heat given to condensing water	K	$G\,(S_k-S_i)$		
Heat supplied to engine	Q	$M\,(XL+S)$	320,050	340,000
Sensible heat at admission	H_0	M_0S_0	2,941	2,878
Internal heat at admission	H_0'	$100\,\frac{V_c+V_0}{C_0}\,I_0$	7,195	7,075
Sensible heat at cut-off	H_1	$(M+M_0)\,S_1$	90,200	93,500
Internal heat at cut-off	H_1'	$100\,\frac{V_c+V_1}{C_1}\,I_1$	181,400	175,500
Sensible heat at release	H_2	$(M+M_0)\,S_2$	68,500	72,900
Internal heat at release	H_2'	$100\,\frac{V_c+V_2}{C_2}\,I_2$	198,000	213,300
Sensible heat, beginning of compression	H_3	M_0S_3	2,277	2,250
Internal heat, beginning of compression	H_3'	$100\,\frac{V_c+V_3}{C_3}\,I_3$	6,475	9,225
Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-\frac{W_a}{778}$	42,876	64,703
Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-\frac{W_b}{778}$	−10,470	−32,920
Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-\frac{W}{778}$	−16,433	−16,065
Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-H_x-H_x'-\frac{Wd}{778}$	−1,107.5	2,170
Heat discharged, and work	B	$K+K'+H_x+H_x'+\frac{W}{778}$	305,184.5	322,113
Jacket and radiation	D	$Q-B$	14,865.5	17,887
Jacket and radiation	D'	$Q_a+Q_b+Q_c+Q_d$	14,865.5	17,887
Quality of steam entering	X	per calorimeter ... per cent.	98.95	98.95
Quality of steam at cut-off	X_1	$100\,\frac{V_c+V_1}{(M+M_0)\,C_1}$ "	81.85	76.20
Quality of steam at release	X_2	$100\,\frac{V_c+V_2}{(M+M_0)\,C_2}$ "	88.90	86.20
Quality of steam in exhaust	X_5	$\left(\frac{K+K'}{M-M_x}-S_5\right)+L_5$ "	88.0	89.3
Heat lost, admission	a	Q_a+Q "	13.41	19.06
Heat restored, expansion	b	Q_b+Q "	−3.266	−9.68
Heat rejected, exhaust	c	Q_c+Q "	−5.13	−4.728
Heat lost, compression	d	Q_d+Q "	−.3461	.688
Heat utilized, work	w	$\frac{W}{778}+Q$ "	5.66	5.40
Heat lost, radiation	r	$D+Q$ "	4.645	5.26
Ratio radiation to work		$r+w$	.819	.977
Ratio cylinder condensation to work		$a+w$	2.359	3.537
Thermodynamic efficiency	E	$(t-t_5)+(461+t)$ ... per cent.	9.175	8.745
Actual efficiency	E_1	$\frac{W}{778}+Q$ "	5.66	5.40
Efficiency compared with ideal	E'	E_1+E "	61.8	61.8

Special symbols V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air pump discharge

Correct for M_x when necessary.

APPLICATION OF HIRN'S ANALYSIS.

INTERMEDIATE.

[FORM No. 1.]

DATA AND RESULTS.

Which engine—intermediate pressure cylinder.
Diameter of cylinder, 48 inches.
Length of stroke, 60 inches.
Diameter of piston rods (2), 4 inches.
Volume of cylinder, crank end, 62.01.
Volume head end, 62.830.
Volume of clearance, cu. ft., head, .942.
Clearance in per cent. of stroke, 1.5.
Volume of clearance, cu. ft., crank, .932.
Clearance in per cent. of stroke, 1.5.
Receiver pressure by gauge, 32.43.
Revolutions per hour, 1,218.8.
Steam used during run, pounds, 162,-524.5.
Weight of condensed steam per hour, 6,771.8.
Lbs. of wet steam per revolution, 5.56.
Per stroke, head, 2.915.
Per stroke, crank, 2.545.

SYMBOLS.

To denote different portions of the stroke, the following subscripts are used: Admission (a); expansion (b); exhaust (c); compression (d).

To denote different events of the stroke, the following sub-numbers are used: Cut-off (1); release (2); compression, beginning of (3); admission, beginning of (0); in exhaust (5).

Quality of steam denoted by x.

Cut-off, crank end per cent. of stroke, 32.85.
Cut-off, head end per cent. of stroke, 35.10.
Compression, crank end per cent. of stroke, 2.92.
Compression, head end per cent. of stroke, 1.67.
Release, crank end, 100.
Release, head end, 100.
I. H. P., head, 88.84.
I. H. P., crank, 80.78.
Total I. H. P., 169.62.

INTERMEDIATE.

[FORM No. 2.]

ABSOLUTE PRESSURES FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

		Cut-Off.	Release.	Beginning.		Symbols. Rankine.	Symbols. Clausius.
				Compression.	Of admission.		
Subscripts used		1	2	3	0		
Absolute pressure	Head	41.5	15.3	16.9	50.1		
	Crank	41.5	14.34	17.1	48.1	P	p
Heat of liquid	Head	238.64	182.59	187.74	250.48		
	Crank	238.64	179.16	188.36	247.88	S	q
Internal latent heat	Head	848.64	892.26	888.24	839.55		
	Crank	848.64	894.68	887.76	841.55	I	ρ
Latent heat evaporation	Head	925.46	964.62	961.03	917.17		
	Crank	925.46	967.04	960.59	918.99	L	r
Total heat	Head	1,164.10	1,147.21	1,148.77	1,167.65		
	Crank	1,164.10	1,146.20	1,148 95	1,166.87	H	λ
Volume 1 lb. cubic feet	Head	9.938	25.09	23.11	8.323		
	Crank	9.938	27.05	22.84	8.549	C	v
Volumes symbols		$V_c + V_1$	$V_c + V_2$	$V_c + V_3$	$V_0 + V_c$	V_c	..
Volumes head, cubic feet		23.01	63.77	2.777	.9425	.9425	..
Volumes crank, cubic feet		21.55	63.77	1.990	.9425	.9425	..

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End.		Crank End.	
		Mean Pressures.	External Work. B. T. U.	Mean Pressures.	External Work. B. T. U.
Symbols		MEP	AW*	MEP	AW*
Admission	a	30.38	123.7	29.25	110.1
Expansion	b	9.92	69.4	8.35	64.15
Exhaust	c	.736	8.41	.265	2.95
Compression	d	3.827	.745	7.19	2.405
Total, net		.....	183.955	.....	168.895

* $A = \frac{1}{778}$. V_c = volume in clearance spaces.

INTERMEDIATE.

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 STROKES.

JACKET STEAM INCLUDED.

Quantities.	Symbols.	Formulæ.	Head.	Crank.
Steam from boiler, pounds	M		291.5	254.5
Steam in clearance, pounds	M_c	$100\,(V_c+C_cX_c)$	11.31	11.02
Steam at admission, pounds	M_0	$100\,\frac{V_c+V_0}{C_0X_0}$	11.31	11.02
Steam, total, pounds	$M+M_0$		302.81	265.52
Heat of condensed steam	K'	MS_g		
Condensing water, pounds	G	$K+K'=$	306,838.5	262,623.5
Heat given to condensing water	K	$G\,(S_k-S_i)$		
Heat supplied to engine	Q	$M\,(XL+S)$	340,100	297,400
Sensible heat at admission	H_0	M_0S_0	2,810	2,734
Internal heat at admission	H_0'	$100\,\frac{V_c+V_0}{C_0}\,I_0$	9,510	9,270
Sensible heat at cut-off	H_1	$(M+M_0)\,S_1$	72,300	63,400
Internal heat at cut-off	H_1'	$100\,\frac{V_c+V_1}{C_1}\,I_1$	197,300	184,100
Sensible heat at release	H_2	$(M+M_0)\,S_2$	55,300	47,600
Internal heat at release	H_2'	$100\,\frac{V_c+V_2}{C_2}\,I_2$	223,300	210,800
Sensible heat, beginning of compression	H_3	M_0S_3	2,122	2,079
Internal heat, beginning of compression	H_3'	$100\,\frac{V_c+V_3}{C_3}\,I_3$	10,670	7,750
Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-\frac{W_a}{778}$	70,450	50,894
Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-\frac{W_b}{778}$	−15,940	−17,315
Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-\frac{W_c}{778}$	−40,189.5	−13,757.5
Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-H_x-H_x'-\frac{W_d}{778}$	546.5	−1,934.5
Heat discharged, and work	B	$K+K'+H_x+H_x'+\frac{W}{778}$	325,233	279,513
Jacket and radiation	D	$Q-B$	14,867.0	17,887
Jacket and radiation	D'	$Q_a+Q_b+Q_c+Q_d$	14,867.0	17,887
Quality of steam at cut-off	X_1	$100\,\frac{V_c+V_1}{(M+M_0)\,C_1}$ ..... per cent.	76.6	81.7
Quality of steam at release	X_2	$100\,\frac{V_c+V_2}{(M+M_0)\,C_2}$ ..... "	83.9	88.9
Quality of steam in exhaust	X_5	$\left(\frac{K+K'}{M-M_x}-S_5\right)+L_5$ .... "	90.25	88.25
Heat lost, admission	a	Q_a+Q ..... "	20.70	17.12
Heat restored, expansion	b	Q_b+Q ..... "	−4.695	−5.83
Heat rejected, exhaust	c	Q_c+Q ..... "	−11.81	−4.628
Heat lost, compression	d	Q_d+Q ..... "	.1608	.6505
Heat utilized, work	w	$\frac{W}{778}+Q$ ..... "	5.41	5.68
Heat lost, radiation	r	$D+Q$ ..... "	4.37	6.015
Ratio radiation to work		$r\div w$	.807	1.059
Ratio cylinder condensation to work		$a\div w$	3.823	3.016
Thermo-dynamic efficiency	E	$(t-t_3)\div(461+t)$ ..... per cent.	9.03	8.95
Actual efficiency	E_1	$\frac{W}{778}\div Q$ ..... "	5.41	5.68
Efficiency compared with ideal	E'	$E_1\div E$ ..... "	60.0	63.53

Special symbols V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air pump discharge.

Correct for M_x when necessary.

LOW PRESSURE.

APPLICATION HIRN'S ANALYSIS.

[FORM No. 1.]

DATA AND RESULTS.

Which engine—low-pressure cylinder.
Diameter of cylinder, 74 inches.
Length of stroke, 60 inches.
Diameter of piston rods (2), 4 inches.
Volume of cylinder, crank end, 148.60.
Volume head end, 149.45.
Volume of clearance, cu. ft., head, 1.150.
Clearance in per cent. of stroke, .77.
Volume of clearance, cu. ft., crank, 1.144.
Clearance in per cent. of stroke, .77.
Receiver pressure by gauge, 1.3.
Barometer, 14.5.
Revolutions per hour, 1,218.8.
Steam used during run, pounds, 162,-524.5.
Weight of condensed steam per hour, 6,771.8.
Per stroke, head, 282.9.
Per stroke, crank, 273.5.
Temperatures of condensed steam, 102.
Temperatures of condensing water, cold, 40.
Hot, 98.
Pounds of condensing water, per hour, 99,600.
Per revolution, 81.8.

SYMBOLS.

To denote different portions of the stroke the following subscripts are used: Admission (a); expansion (b); exhaust (c); compression (d).

To denote different events of the stroke, the following sub-numbers are used: Cut-off (1); release (2); compression, beginning of (3); admission, beginning of (0); in exhaust (5).

Quality of steam denoted by x.

Cut-off, crank end per cent. of stroke, 39.1.
Cut-off, head end per cent. of stroke, 38.3.
Compression, crank end per cent. of stroke .64.
Compression, head end per cent. of stroke, 3.13.
I. H. P., head, 116.23.
I. H. P., crank, 112.63.
Total I. H. P., 228.86.
Release, crank end, 98.8.
Release, head end, 98.9.

LOW PRESSURE.

[FORM No. 2.]

ABSOLUTE PRESSURE FROM INDICATOR DIAGRAMS AND CORRESPONDING PROPERTIES OF SATURATED STEAM.

		Cut-Off.	Release.	Beginning.		Symbols. Rankine.	Symbols. Clausius.
				Compression.	Of admission.		
Subscripts used		1	2	3	0		
Absolute pressure	Head	13.3	5.3	1.8	16.2	P	p
	Crank	13.9	5.3	1.5	16.5		
Heat of liquid	Head	175.51	132.93	89.48	185.54	S	q
	Crank	187.77	132.93	82.23	186.48		
Internal latent heat	Head	897.81	931.37	963.80	889.96	I	ρ
	Crank	896.09	951.37	971.68	889.23		
Latent heat evaporation	Head	969.60	999.26	1029.47	962.56	L	r
	Crank	868.01	999.26	1034.55	961.91		
Total heat	Head	1145.11	1132.19	1118.95	1148.11	H	λ
	Crank	1145.78	1132.19	1116.77	1148.39		
Volume 1 lb. cubic feet	Head	28.96	69.24	202.5	240.06	C	v
	Crank	27.78	69.24	251.1	23.66		
Volumes symbols		$V_c + V_1$	$V_c + V_2$	$V_c + V_3$	$V_0 + V_c$	V_c	..
Volumes head, cubic feet		58.349	148.7	5.824	1.149	1.149	..
Volumes crank, cubic feet		59.949	148.7	2.105	1.149	1.149	..

MEAN PRESSURES AND HEAT EQUIVALENTS OF EXTERNAL WORK.

	Subscripts.	Head End.		Crank End.	
		Mean Pressures.	External Work. B. T. U.	Mean Pressures.	External Work. B. T. U.
Symbols		MEP	AW*	MEP	AW*
Admission	a	.5613	5.95	.713	7.64
Expansion	b	−6.33	−108.54	−6.38	−106.6
Exhaust	c	−12.66	−338.2	−12.19	−332.6
Compression	d	−9.77	−8.515	−7.548	−1.33
Total	.....	.....	244.125	.....	234.97

* $A = \frac{1}{778}$. V_c = volume in clearance spaces.

LOW PRESSURE.

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 STROKES.

JACKET STEAM INCLUDED.

Quantities.	Symbols.	Formulæ.	Head.	Crank.
Steam from boiler, pounds	M		282.9	273.5
Steam in clearance, pounds	M_c	$100\,(V_c + C_c X_c)$	4.76	4.85
Steam at admission, pounds	M_0	$100\,\frac{V + V_0}{C_0 X_0}$	4.76	4.85
Steam, total, pounds	$M + M_0$		287.66	278.35
Heat of condensed steam	K'	$M S_g$	19,040	19,880
Condensing water, pounds	G		4,002	4,182
Heat given to condensing water	K	$G\,(S_k - S_l)$	232,100	242,400
Heat supplied to engine	Q	$M\,(XL + S)$	324,600	314,000
Sensible heat at admission	H_0	$M_0 S_0$	835	911
Internal heat at admission	H_0'	$100\,\frac{V_c + V_0}{C_0} I_0$	4,280	4,310
Sensible heat at cut-off	H_1	$(M + M_0)\,S_1$	50,450	52,350
Internal heat at cut-off	H_1'	$100\,\frac{V_c + V_1}{C_1} I_1$	160,500	198,100
Sensible heat at release	H_2	$(M + M_0)\,S_2$	38,180	37,000
Internal heat at release	H_2'	$100\,\frac{V_c + V_2}{C_2} I_2$	199,900	199,900
Sensible heat, beginning of compression	H_3	$M_0 S_3$	425.8	398.4
Internal heat, beginning of compression	H_3'	$100\,\frac{V_c + V_3}{C_3} I_3$	2,778	816
Cylinder loss during admission	Q_a	$Q + H_0 + H_0' - H_1 - H_1' - \frac{W_a}{778}$	118,160	73,007
Cylinder loss during expansion	Q_b	$H_1 + H_1' - H_2 - H_2' - \frac{W_b}{778}$	−16,276	19,210
Cylinder loss during exhaust	Q_c	$H_2 + H_2' - H_3 - H_3' - K - K' - \frac{W_c}{778}$	−50,083.8	−59 854.4
Cylinder loss during compression	Q_d	$H_3 + H_3' - H_0 - H_0' - H_x - H_x' - \frac{W_d}{778}$	−2,752.7	−4,139.6
Heat discharged, and work	B	$K + K' + H_x + H_x' + \frac{W}{778}$	275,552.5	285,777
Jacket and radiation	D	$Q - B$	49,047.5	28,223
Jacket and radiation	D'	$Q_a + Q_b + Q_c + Q_d$	49,047.5	28,223
Quality of steam at cut-off	X_1	$100\,\frac{V_c + V_1}{(M + M_0)\,C_1}$ per cent.	70.2	77.5
Quality of steam at release	X_2	$100\,\frac{V_c + V_2}{(M + M_0)\,C_2}$ " "	74.6	77.1
Quality of steam in exhaust	X_5	$\left(\frac{K + K'}{M - M_x} - S_5\right) + L_5$ " "	77.8	83.3
Heat lost, admission	a	$Q_a + Q$ " "	36.42	23.25
Heat restored, expansion	b	$Q_b + Q$ " "	−5.008	6.120
Heat rejected, exhaust	c	$Q_c + Q$ " "	−15.40	−19.10
Heat lost, compression	d	$Q_d + Q$ " "	.848	1.308
Heat utilized, work	w	$\frac{W}{778} + Q$ " "	7.525	7.485
Heat lost, radiation	r	$D + Q$ " "	15.08	8.99
Ratio, radiation to work		$r + w$	1.995	1.201
Ratio, cylinder condensation to work		$a + w$	4.84	2.977
Thermo-dynamic efficiency	E	$(t - t_5) \div (461 + t)$ per cent.	11.28	10.70
Actual efficiency	E_1	$\frac{W}{778} \div Q$ " "	7.525	7.485
Efficiency compared with ideal	E'	$E_1 \div E$ " "	67.8	70.01

Special symbols V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge.

Correct for M_x when necessary.

APPENDIX D.

COMPLETE THERMAL ANALYSIS.

JACKET-HEAT, RADIATION LOSSES, AND HEAT REJECTED FROM EACH CYLINDER PER 100 REVOLUTIONS.

Total heat admitted to high-pressure cylinder.....	598,800	B. T. U.
Total heat used in all jackets, assuming ⅓ weight of jacket steam to be used in each jacket.....	40,870	"
Total heat used	639,670	"
Heat rejected from low-pressure cylinder..... ...	513,420	"
Total work done.....	119,652	"
Total radiation loss.....	6,598	"
	639,670	"
Radiation loss from each cylinder assumed to be equal in all cylinders.....	2,199	"

Heat rejected from high-pressure cylinder = heat entering + heat supplied by jacket − radiation loss − work done

$$= 598{,}800 + 13{,}360 - 2{,}199 - 36{,}456$$
$$= 573{,}505 \text{ B. T. U.}$$

Heat rejected from intermediate-pressure cylinder = heat rejected from high-pressure cylinder + heat supplied by jacket − radiation loss − work done

$$= 573{,}505 + 13{,}360 - 2{,}199 - 35{,}286$$
$$= 549{,}380 \text{ B. T. U.}$$

Heat rejected from low-pressure cylinder = heat rejected from intermediate-pressure cylinder + heat supplied by jacket − radiation loss − work done

$$= 549{,}380 + 14{,}150 - 2{,}200 - 47{,}910$$
$$= 513{,}420 \text{ B. T. U.}$$
$$= K + K' \text{ (see low-pressure analysis).}$$

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS. HIGH-PRESSURE CYLINDER (JACKET STEAM EXCLUDED) WITH FIRST RECEIVER.

Quantities.	Symbols.	Formulæ.	
1. Steam from boiler entering working cylinder, pounds	M	.....	504.5
2. Steam in clearance, pounds	M_c	$100\,(V_c+C_cX_c)$	18.158
3. Steam at admission, pounds	M_0	$100\,(V_c+V_0+v_0$	.....
4. Steam used by calorimeter, pounds	.....	.....	.....
5. Steam, total, pounds	$M+M_0$	.....	522.658
6. Heat of condensed steam	K'	Mq_g	.....
7. Condensed water, pounds	G	.....	.....
8. Heat given to condensing water	K	$G\,(q_k-q_l)$ (Heat rejected.)	573,505
9. Heat supplied to engine	Q	$M\,(xr+q)$	598,800
10. Sensible heat at admission	H_0	M_0q_0	5,819
11. Internal heat at admission	H_0'	$100\,\frac{V_c+V_0}{v_0}\,\rho_0$	14,270
12. Sensible heat at cut-off	H_1	$(M+M_0)\,q_1$	167,400
13. Internal heat at cut-off	H_1'	$100\,\frac{V_c+V_2}{v_1}\,\rho_1$	356,900
14. Sensible heat at release	H_2	$(M+M_0)\,q_2$	128,700
15. Internal heat at release	H_2'	$100\,\frac{V_c+V_2}{v_2}\,\rho_2$	411,300
16. Sensible heat, beginning of compression	H_3	M_0q_3	4,527
17. Internal heat, beginning of compression	H_3'	$100\,\frac{V_c+V_3}{v_3}\,\rho_3$	15,700
18. Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-AW_a$	+62.629
19. Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-AW_b$	−46.990
20. Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-AW_c$	−27,862
21. Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-AW_d$	−1,064
22. Heat discharged, and work	B	$K+K'+AW$	+609,961
*23. Jacket and Loss	D	$Q-B$	−11,161
*24. Jacket and Loss	D'	$Q_a+Q_b+Q_c+Q_d$	−11,161
25. Quality of steam entering	x	per calorimeter ..... per cent.	98.95
26. Quality of steam at cut-off	x_2	$100\,\frac{V_cV_1}{(M+M_0)\,v_1}$ ..... "	86.77
27. Quality of steam at release	x_2	$100\,\frac{V_c+V_2}{(M+M_0)\,v_2}$ ..... "	93.3
28. Quality of steam at compression	.....	$100\,\frac{V_c+V_3}{M_0v_3}$ ..... "	.....
29. Quality of steam at admission	x_0	per calorimeter ..... "	.....
30. Quality of steam in exhaust	x_5	$\left(\frac{K+K'}{M-M_x}-q_5\right)\div r_5$ ..... "	96.8
31. Heat lost, admission	a	$Q_a\div Q$ ..... "	10.46
32. Heat restored, expansion	b	$Q_b\div Q$ ..... "	−7.86
33. Heat rejected, exhaust	c	$Q_c\div Q$ ..... "	−4.65
34. Heat lost, compression	d	$Q_d\div Q$ ..... "	.18
35. Heat utilized, work	w	$\frac{W}{778}\div Q$ ..... "	6.087
36. Heat lost, radiation	R	radiation$\div Q$ ..... "	367
37. Ratio, radiation to work	.....	$R\div w$	.0603
38. Ratio, cylinder condensation to work	.....	$a\div w$	1.72
39. Thermo-dynamic efficiency	E	$(t-t_5)\div(460+t)$ ..... per cent.	8.96
40. Actual efficiency	E_1	$AW\div Q$ ..... "	6.087
41. Efficiency compared with ideal	E'	$E_1\div E$ ..... "	68.0

Special symbols V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A=\frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* This quantity is the difference between the heat lost by radiation and that received from the jacket. The negative sign shows that more heat is received than lost.

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS—INTERMEDIATE CYLINDER (JACKET STEAM EXCLUDED) WITH SECOND RECEIVER.

Quantities.	Symbols.	Formulæ.		
1. Steam entering cylinder, pounds	M			504.5
2. Steam in clearance, pounds	M_c	$100\,(V_c+V_0)\div v_0$		22.33
3. Steam used by calorimeter, pounds				
4. Steam, total, pounds	$M+M_0$			526.83
5. Heat of condensed steam	K'	Mq_g		
6. Heat rejected	G			549,380
7. Heat given to condensing water	K	$G\,(q_k-q_i)$		
8. Heat supplied to engine	Q	$M(xr+q)$		573,505
9. Sensible heat at admission	H_0	M_0q_0		5,534
10. Internal heat at admission	H_0'	$100\,\frac{V_c+V_0}{v_0}\,\rho_0$		18,780
11. Sensible heat at cut-off	H_1	$(M+M_0)\,q_1$		125,600
12. Internal heat at cut-off	H_1'	$100\,\frac{V_c+V_2}{v_1}\,\rho_1$		381,400
13. Sensible heat at release	H_2	$(M+M_0)\,q_2$		95,300
14. Internal heat at release	H_2'	$100\,\frac{V_c+V_2}{v_2}\,\rho_2$		434,100
15. Sensible heat, beginning of compression	H_3	M_0q_3		4,201
16. Internal heat, beginning of compression	H_3'	$100\,\frac{V_c+V_2}{v_3}\,\rho_3$		18,420
17. Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-A\,W_a$		67,439
18. Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-A\,W_b$		−35,755
19. Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-A\,W_c$		−41,465
20. Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-A\,W_d$		−1,380
21. Heat discharged, and work	B	$K+K'+A\,W$		584,666
*22. Jacket and Loss	D	$Q-B$		−11,161
*23. Jacket and Loss	D'	$Q_a+Q_b+Q_c+Q_d$		−11,161
24. Quality of steam entering	x	per calorimeter	per cent.	
25. Quality of steam at cut-off	x_2	$100\,\frac{V_cV_1}{(M+M_0)\,v_1}$	"	85.4
26. Quality of steam at release	x_2	$100\,\frac{V_c+V_2}{(M+M_0)\,v_2}$	"	93.2
27. Quality of steam at compression		$100\,\frac{V_c+V_3}{M_0v_3}$	"	
28. Quality of steam at admission	x_0	per calorimeter	"	
29. Quality of steam in exhaust	x_5	$\left(\frac{K+K'}{M-M_x}-q_5\right)\div r_5$	"	94.0
30. Heat lost, admission	a	$Q_a\div Q$	"	11.75
31. Heat restored, expansion	b	$Q_b\div Q$	"	−6.23
32. Heat rejected, exhaust	c	$Q_c\div Q$	"	−7.23
33. Heat lost, compression	d	$Q_d\div Q$	"	−0.24
34. Heat utilized, work	w	$\frac{W}{778}\div Q$	"	6.155
35. Heat lost, radiation	R	Radiation $\div\,Q$	"	.384
36. Ratio, radiation to work		$R\div w$		.0623
37. Ratio, cylinder condensation to work		$a\div w$		1.795
38. Thermo-dynamic efficiency	E	$(t-t_5)\div(460+t)$	per cent.	8.99
39. Actual efficiency	E_1	$A\,W\div Q$	"	6.155
40. Efficiency compared with ideal	E'	$E_1\div E$	"	68.55

Special symbols, V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A=\frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* Same explanation as for high-pressure cylinder. Radiation assumed equal for all three cylinders, and heat received from jackets assumed equal in high and intermediate pressure cylinders.

HIRN'S ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS—LOW-PRESSURE CYLINDER (JACKET STEAM EXCLUDED).

Quantities.	Symbols.	Formulæ.	
1. Steam entering working cylinder, pounds	M	....	504.5
2. Steam in clearance, pounds	M_c	$100(V_c+V_0)\div v_0$	9.61
3. Steam used by calorimeter, pounds	...	...	...
4. Steam, total, pounds	$M+M_0$	...	514.11
5. Heat of condensed steam	K'	Mq_g	38,920
6. Condensed water, pounds	G	...	8,184
7. Heat given to condensing water	K	$G(q_k-q_l)$	474,500
8. Heat supplied to engine	Q	$M(xr+q)$	549,380
9. Sensible heat at admission	H_0	M_0q_0	1,746
10. Internal heat at admission	H_0'	$100\frac{V_c+V_0}{v_0}\rho_0$	8,590
11. Sensible heat at cut-off	H_1	$(M+M_0)q_1$	93,450
12. Internal heat at cut-off	H_1'	$100\frac{V_c+V_2}{v_1}\rho_1$	353,600
13. Sensible heat at release	H_2	$(M+M_0)q_2$	68,300
14. Internal heat at release	H_2'	$100\frac{V_c+V_2}{v_2}\rho_2$	399,800
15. Sensible heat, beginning of compression	H_3	M_0q_3	824.2
16. Internal heat, beginning of compression	H_3'	$100\frac{V_c+V_3}{v_3}\rho_3$	3,594
17. Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-AW_a$	11,307
18. Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-AW_b$	464
19. Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-AW_c$	−116,818.2
20. Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-AW_d$	−6,902.8
21. Heat discharged, and work	B	$K+K'+AW$	561,330
*22. Jacket and Loss	D	$Q-B$	−11,950
*23. Jacket and Loss	D'	$Q_a+Q_b+Q_c+Q_d$	−11,950
24. Quality of steam entering	x	per calorimeter ..... per cent.	.....
25. Quality of steam at cut-off	x_2	$100\frac{V_cV_1}{(M+M_0)v_1}$ ..... "	81.15
26. Quality of steam at release	x_2	$100\frac{V_c+V}{(M+M_0)v_2}$ ..... "	83.33
27. Quality of steam at compression	...	$100\frac{V_c+V_3}{M_0v_3}$ ..... "	...
28. Quality of steam at admission	x_0	per calorimeter. ..... "	...
29. Quality of steam in exhaust	x_5	$\left(\frac{K+K'}{M-M_x}-q_5\right)\div r_5$ ..... "	89.95
30. Heat lost, admission	a	$Q_a\div Q$ ..... "	2.056
31. Heat restored, expansion	b	$Q_b\div Q$ ..... "	.084
32. Heat rejected, exhaust	c	$Q_c\div Q$ ..... "	−21.22
33. Heat lost, compression	d	$Q_d\div Q$ ..... "	−1.256
34. Heat utilized, work	w	$\frac{W}{778}\div Q$ ..... "	8.7
35. Heat lost, radiation	R	Radiation $\div Q$ ..... "	.40
36. Ratio, radiation to work	...	$R\div w$	.046
37. Ratio, cylinder condensation to work	...	$a\div w$	.236
38. Thermo-dynamic efficiency	E	$(t-t_3)\div(460+t)$ ..... per cent.	10.98
39. Actual efficiency	E_1	$AW\div Q$ ..... "	8.70
40. Efficiency compared with ideal	E'	$E_1\div E$ ..... "	79.18

Special symbols, V_c = volume clearance, = t measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A=\frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* Same explanation as for high pressure cylinder.

DISCUSSION.

Prof. C. H. Peabody.—The test of the triple expansion engine at North Point Station, Milwaukee, conducted by Professor Carpenter and reported here by Professor Thurston, is of the greatest importance to all who are interested in steam engineering. The designer and builders of the engine may well be satisfied with a report showing the best performance as yet attained on any engine. There are, however, some things in the paper to which I am inclined to take exception.

It appears to me an error to report the *dry* steam per horse-power per hour. The engine actually used 11.8 pounds of steam per horse-power per hour; a sufficiently remarkable result. The steam carried about one per cent. of moisture; if allowance is made for that moisture, the dry steam used appears to be one-tenth of a pound less. The errors of the determination of the indicated horse-power have at least as great a magnitude as this. We may, therefore, consider the two figures as substantially identical; and, to my mind, the making of *any* allowance more than offsets the slight reduction thus obtained. Now we find, by the aid of the recent types of calorimeters, that steam supplied to engines in good practice seldom contains less than half a per cent., or more than one and a half per cent., of moisture. The uncertainty due to priming is, then, not greater than the variation in the value of a pound of steam at different pressures, and is less than the allowance to be made for the temperature of the feed-water; if some of the steam is used in steam-jackets, the case becomes very much involved. It appears to me desirable that the results of an engine test should give the actual quantity of steam used, together with the steam pressure, the priming, the temperature of the feed-water, and if jackets are used, the steam condensed therein; all these and other important data are customarily reported now and have been for many years. If one wishes an exact statement of the consumption of the engine, it can be had in the form of thermal units per horse-power per minute, as has already become customary.

There is another point which appears to me even more objectionable; namely, the assigning a number of pounds of steam per horse-power per hour to an ideal engine working on Carnot's

cycle. The actual steam engine admits steam to, and exhausts steam from, the cylinder; and *uses* some number of pounds of steam per horse-power per hour. We may readily calculate the steam that would be *used* by an engine without mechanical imperfections, and with its cylinder made of non-conducting materials; such an engine is as much an ideal engine as one using Carnot's cycle, but the steam used is a real mathematical quantity. It is quite different with the engine using Carnot's cycle; nothing enters or leaves the cylinder, but we have isothermal and adiabatic expansions and compressions, during which steam is formed or is condensed. It is only by a mathematical fiction that we can assign a steam consumption to that engine, and such a mathematical fiction leads only to confusion and misconceptions. There is one thing, and one thing only, that is of practical value connected with Carnot's cycle; namely, it allows us to calculate the maximum efficiency of an engine working through a given range of temperatures. The comparison of the efficiency of the actual engine, with the efficiency of the non-conducting engine which has a similar cycle, and with the efficiency of Carnot's cycle, gives us real and valuable information; the comparison of the heat consumption of these three cycles, in thermal units, gives exactly the same information. But the comparison of the steam consumption of the actual engine, with the steam consumption assigned by the aforesaid mathematical fiction to Carnot's cycle, gives a result which is inconsistent with that from the legitimate comparison; such a method is therefore misleading and unwarranted.

Before proceeding with this same subject, I would like to ask how the efficiency of this engine, given as 0.18727 on page 338, was obtained. Using the heat consumption 217.6 B. T. U., on page 338, in the equation on page 348 we get

$$\text{Eff.} = \frac{42.42}{217.6} = 0.194.$$

Again, on page 357, the author refers to "the common—the Rankine cycle." I wish he would kindly say just what he means here. Rankine in *The Steam Engine* discusses two cases—one for jacketed engines and one for unjacketed engines. May we assume that the case of jacketed engines is intended, from the following quotation from page 358? "The case here studied is

that of the jacketed engine, and it is assumed, with Rankine, that the jacket has precisely that effectiveness of action which insures the retention of the working charge in the dry and saturated state up to the point of rejection with the opening of the exhaust valve." If so, we have but to turn to page 376 and following to see that that is just what the jackets do not do. The following table, extracted from the Hirn's analysis, tells the story:

QUALITY OF STEAM IN THE CYLINDERS, PER CENT.

	High Pressure.		Intermediate.		Low Pressure.	
	H. E.	C. E.	H. E.	C. E.	H. E.	C. E.
Cut-off	81.85	86.77 76.20 93.3	76.6	85.4 81.7 93.2	70.2	81.15 77.5 83.3
Release	83.90	86.20	83.9	83.9	74.6	77.1

It needs but to add that the steam supplied had one per cent. of moisture, and that the quality of steam in the exhaust from the low-pressure cylinder was 84.3%. As the steam became progressively wetter on the way through the engine, the steam supplied to the intermediate and low pressure cylinders must have had a quality between 99% and 84%.

In view of this, and of the formidable array of tables and diagrams occupying the latter half of the paper, I am led to protest that there is but one efficiency for the steam-engine; namely, the ratio of the heat changed into work to the heat supplied by the boiler. There is some interest and advantage in comparing this efficiency with that of a non-conducting engine and with Carnot's cycle. This shows us what we are doing, what we are trying to do, and what is the limit of possibility, and would appear to exhaust the subject.

I thought it would be of some interest to calculate the three efficiencies and compare them with results from the engine in the laboratory of the Massachusetts Institute of Technology; so I have prepared the following table:

	Engines at	
	North Point.	M. I. T.
Sizes of cylinders	28, 48, 76–60	9, 16, 24–30
Steam pressures	135.94	157.7
Pressures at release	5.3	6.0
Mean back pressure	2.0	4.5
Steam per horse-power per hour	11.8	13.7
Efficiency of engine	0.194	0.183
" " non-conducting engine	0.235	0.225
" " Carnot's cycle	0.276	0.245

Attention should be called to the fact that, of the diagram on page 353, the first two divisions on the left represent our present experimental knowledge of the properties of saturated steam; the other fifty-eight divisions would appear to represent rather a large extension of the curves into the region of the unknown.

On page 367 an attempt is made to deduce the saving from the use of the steam jackets, from considerations of cylinder condensation, adiabatic expansion, etc. With an engine showing so remarkable an economy, one must hesitate to say whether any change could improve it, and it appears that the jackets were well designed and applied; but there is but one way of knowing how much is gained by the use of the jackets, or whether there is any gain in this case, and that is to make the proper tests with and without steam in the jackets.

Finally, I desire to disclaim any responsibility for the publication of the diagrams for the action of the reciprocating parts on pages 324 and 325. The work was done by one of my assistants, and a tracing was sent to the builders of the engine in acknowledgment of their courtesy in furnishing information. I was surprised to find a reproduction of the diagrams in their report of this same test, and was more surprised to find it honored by an appearance in the *Transactions* of our society. I could have wished for an opportunity to revise the work, to make some changes, and to offer some explanation of its meaning. If it can be done, I would like to have the figure on page 324 follow that on page 325; the two really ought to be on pages facing each other. If that is done, the work will be consistent; I believe it to be substantially correct.

Mr. George I. Rockwood.—Appreciating the large expenditure of time and labor attending the preparation of Professor Thurston's interesting paper, and the comparative ease with which cavil-

ling criticism may be offered, I should be more disposed to praise its thoroughness rather than to criticise, were it not for the fact that the main figure—the figure which makes the test of remarkable interest—namely, $11\frac{678}{1000}$ lbs. steam used per I. H. P. per hour, is adduced by the author (on pages 366, 367) as proof of the correctness of his previously published "Philosophy of the Multiple-Cylinder Engine." I think it possible to show—both by means of data recently obtained in tests of a two-cylinder compound engine, designed in accordance with a system of compounding devised by myself, and by means of figures given in the paper itself—that the low steam consumption mentioned does not constitute proof of that philosophy, and that the accuracy of the feed-water measurement is at least open to question.

My idea, in thinking that the water test is in error, is based on the following considerations:

1. There is no good reason apparent, *à priori*, why this engine should have been expected by its designers to exceed in economy, as it does by at least 10%, all other engines of its own type "under similar external working conditions."

It is not essentially different, in any particular of its design, from other triple engines furnished by the same builders for other than pumping purposes; nor is it essentially different from Corliss pumping engines built by others.* Indeed, its low steam pressure and rotative speed should have made it less rather than more economical than these other engines of its class.

2. The feed-water test shows less heat consumed than does the heat-unit test proper. In other words, taking the data of the test made April 7 (given on page 337), and even allowing for the greater horse-power, due to an increase of one-tenth of a revolution per minute over the speed existing during the test of March 25, the thermo-dynamic equation which should exist between the amount of heat entering the engine, both through its jackets and through its main steam-pipe, and the amount independently measured as passing out of the engine in the exhaust steam, as radiation and as work, cannot be established excepting on the supposition that more water was actually evaporated in the boilers than the amount as given.

* *Vide* test Narragansett Electric Light Engine or test of Laketon Pumping Engine by Professor Denton.

3. According to the figures given in the table of water per I. H. P. per hour, computed from the diagrams (page 375), there was present in the high-pressure cylinder, just before release, $10\frac{45}{100}$ lbs. of the total $11\frac{678}{1000}$ lbs. The jacket-water, being 9.25% of the total, amounted to 1.08 lbs. This added to $10\frac{45}{100}$ gives $11\frac{53}{100}$ lbs., which, subtracted from $11\frac{678}{1000}$, leaves present in the high-pressure cylinder at release but $\frac{148}{1000}$ of a pound, which is but 1% of the alleged total feed-water. If the figure $11\frac{678}{1000}$ is correct, then here is the secret of the remarkable result. This engine works without cylinder condensation in either of the first two cylinders. It may, however, be set down as the only engine that ever did or ever will operate in that manner, "under similar external working conditions."

4. It seems to be characteristic of the various Western pumping plants having Reynolds Corliss pumps, that the boilers, though designed to be first class in all respects, of course, and though worked with good grades of fuel, invariably do poorly. Why is it? Under the conditions, they should be discarded if they cannot be made to do at least 10% better. But perhaps they are really evaporating this amount more than they are accredited with in the paper. A poor boiler performance and a superlatively fine engine performance would be the result of incorrect water measurement.

5. I am in doubt as to the accuracy of the sketch annexed to page 332. This drawing seems to show that all three weighing barrels are rigidly connected to the same delivery pipe. This pipe was perhaps 2½″ diameter, and therefore large enough to help support or depress one or more of the barrels at any given time. If this rigid pipe connection actually existed at the time of the test, then we need look no further for an explanation of the figure $11\frac{678}{1000}$.

I have indicated the inconsistency, as it seems to me, of certain facts as given in the paper, and the improbability, in general, of such an engine, working under so low a steam pressure and slow a speed, giving the horse-power on so low a steam consumption.

The second part of my critique has to do with the statement of Dr. Thurston, on pages 366, 367, to the effect that the more cylinders adopted with a stated boiler pressure, the less will be the loss by cylinder condensation.

In the volume of *Transactions* for 1892 will be found a paper

containing an account of some compound and triple engine tests, which of themselves would seem to disprove the statement. However that may be, I would like to call the attention of the practical men here present to data from a Rockwood System Compound Engine, built by the Wheelock Engine Company, of Worcester, and running the cotton-mill of the West Boylston Manufacturing Company, at Oakdale, Mass. The cylinders are two in number, and connected as a pair on either side of a 20-foot fly-wheel. Their diameters are respectively $\frac{16\frac{7}{32}''}{40\frac{1}{4}''} \times 48''$ stroke of pistons. An intermediate superheating receiver of large bulk is placed below the floor and between the cylinders. The low-pressure cylinder exhausts through a 14″ exhaust pipe into a Wheelock Adjustable Syphon Condenser; hence, the engine has no air-pump. Steam is supplied by a battery of three Manning upright boilers, situated about 100 feet from the throttle-valve. The boilers are of standard make, for withstanding a pressure of 150 lbs., and have a proportion of total heating surface to grate surface of $\frac{50}{1}$. The boiler-feed duplex pump is driven by steam taken from these boilers.

Just before Thanksgiving Day the plant was twice tested; once with a load of 378 I.H.P., and the next day a load of 484½ I.H.P. The first test was preliminary to the second, which was the final test. The coal and the power were the two quantities measured. Incidentally I also measured the moisture in the steam. To measure the power, at my request, I was kindly loaned four new Tabor Indicators by the manufacturers of them. The coal was weighed by the superintendent of the cotton mill. Following are the results:

TEST WITH LOAD OF 378 I.H.P.

Duration of test	5 hours.
Total coal burned (including that used at banking the fires and to run the feed-pump)	2664 lbs.
Revolutions per minute	77.3
Steam pressure, average	138.5 lbs.
Vacuum by gauge, average	26⅝ inches.
Coal per I. H. P. per hour (without deductions for moisture, ash, or any kind whatsoever)	1.41 lbs.
The number of expansions during this test was	40

TEST WITH A LOAD OF 484½ I.H.P.

Duration of test	9½ hours.
Steam pressure, average	144.2 lbs.
Vacuum by gauge	27½ inches,
Total number of expansions	25
Total coal burned (including that used at noon for banking and that used to run pump)	6358 lbs.
Coal per I. H.P. per hour (without deductions for moisture, ash, or any other kind)	1.38 lbs.

This includes, besides 9½ hours of regular running, a short run of ten minutes just before 1 o'clock, to warm up the engine and get the load well on by then. No coal allowance was made for the slight amount of steam taken from the boilers at this time.

In each test the calorimeter showed moisture in the steam to the amount of 1.6%. But as, when the engine was not running, the instrument still showed 1% of moisture, I consider the actual amount of entrained water to have been a little over one-half of 1%.

Some interesting comparisons may be drawn between the performance of this plant and that reported by Dr. Thurston.

Three Manning upright boilers are here used, while in Milwaukee there were four horizontal boilers. Dr. Thurston ascribes the poor duty of these boilers to "probably" too thin fires. Now, notwithstanding the fact that the rates of combustion and of evaporation in the Manning boilers were nearly identical with those of the horizontal boilers, I had no difficulty in carrying heavy fires. Dr. Thurston remarks that the moisture in the steam was 1.05%, although the boilers are shown in the draught to be quite near the engine, and, as before stated, ebullition in the boilers was quite moderate. I should like to know whether any correction has been made for the moisture indicated by the calorimeter, when the steam was known to be dry.

I leave the data of the compound-engine test as just given, without attempting to figure out of account the effect of various small percentages, such as moisture and ash in coal, moisture in steam, leakage from the boilers, coal used for banking, etc., as the owner of the plant cannot take them into account in settling his coal bills. I may say that I have yet to learn of a single authentic case of a triple-expansion or quadruple-expansion engine which is daily operated with an expenditure of less than 1.4 lbs. of coal per I. H. P. per hour, no deductions being made

for the several small items stated. Not until this duty is shown to be surpassed in regular every-day operation of the plant, can the mill-owner be justified in sinking his money in a multiple-cylinder engine in preference to a two-cylinder compound, properly designed.

I will add to this discussion the indicator diagrams taken during tests, showing the action of the valve-gear and the degrees of expansion in the two tests.

I would say that the scales were sealed by the Sealer of Weights and Measures before being used for weighing the coal, and that Mr. Wm. B. Fittz, superintendent of the Oakdale Mills, was extremely careful in his coal measurements, as we both doubted our first test, on account of the remarkable economy shown; he also figured the horse-power after me, to obtain a check. The indicators were new, and directly from the shops.

A water test, wholly without the connivance of those conducting the trial, may be entirely wrong without their knowledge. If a coal test is used as the basis of one's conclusions as to the economical operation of a steam-engine, it is next to impossible to be mistaken on the score of using *too small* an amount of coal to the horse-power, though confusion reigns, of course, when more than a reasonable amount is consumed. This is not said to disparage water tests, where these can be accurately made. We used George's Creek Cumberland coal, having small ash or moisture.

To deal somewhat further with the facts regarding this engine test which I report, and which Professor Denton has commented upon unfavorably, for not including a feed- and jacket-water test also, I wish to say that I weighed the water that went to the boilers; but, unfortunately, I also weighed much more.

After leaving the weighing barrels, the water went into an open heater and thence (we supposed) to the boilers. On examination it proved that more than one-fourth of the whole amount was wasted by being syphoned out of the overflow pipe. Prof. G. I. Alden, of Worcester Polytechnic School, is intending to carry out a course of experiments on this engine, and I assure you neither I nor any one else interested directly in the performance of this engine will be present.

Besides this engine there are others which will be tested soon.

Among these is one having cylinders of the following dimension, $\frac{18''}{48''} \times 48''$, to work with a boiler-pressure of 160 lbs. Another one of this type has cylinders $\frac{18''}{44''} \times 48''$, steam of 150 lbs. A third is the Lowell pumping engine, designed twenty years ago by Mr. E. D. Leavitt. The old cylinders were removed, and their places supplied with cylinders as follows: high-pressure cylinder, $20'' \times 60''$; low-pressure cylinder, $41'' \times 96''$. This engine formerly worked with a boiler-pressure of only 50 lbs. With the new arrangement the boiler-pressure is 160 lbs. As this engine runs but 13 R.P.M., I do not expect so low a steam-consumption as from some of the others.

This method of compounding has come to stay, I think, because I am getting wonderfully fine results in point of economy without any practical difficulties in operation. Even if the economy in steam were not quite so good as in multiple-expansion engines, there are sufficient practical reasons for adopting this method of compounding in preference to any other. But I am very far from admitting that, given an equal number of expansions and an equal boiler-pressure, it is any less economical than any multiple-cylinder engine whatsoever.

I would also say that the commercial interests of the day frequently require engines to take care of great variations of load. This type of engine is especially fitted to do its work economically under such conditions. There seems to be something about the atmosphere of Milwaukee on account of which engines coming from that city are bound to give extraordinarily good economy. We are having an engine built in one of its engine shops which is to drive electrical machinery for a street railway. This engine also is to be tested.

Mr. F. A. Scheffler.—I wish to call attention to the table on page 350, which is entitled "Steam per I.H.P. per hour at Maximum Efficiency, as above," which, notwithstanding the reference "as above," is extremely misleading. I cannot understand why Dr. Thurston should even have considered the necessary work involved to present such a table, as he, as well as every one else who has given the subject even a small amount of consideration, is aware that to say virtually, even theoretically, if feed-water at 110° temperature is evaporated into steam at 90 lbs. pressure, and used with the highest efficiency theoretically ob-

tainable, that the consumption of 8.68 lbs. of steam will be used per I.H.P. per hour, and that if the feed-water temperature is increased 100° to 210°, under the same conditions, that it will require more than twice as much steam per I.H.P. per hour, or 18.02 lbs. If this theory is correct, and we all know that practically it is absolutely not correct, why not deliver the feed-water to the boiler at 32°, or even at zero, and use one-half or one-fourth as much as would be required at 110°?

This Society is after facts as well as theoretical data, and I can see no good cause for the lot of calculations, with results, which are presented in this particular table. The title of the paper is very exact in its meaning of what is intended to be covered in the paper. There is, however, no "contemporary economy of the steam-engine" that I can find or ever expect to find in this table. Perhaps I do not fully comprehend the idea in Dr. Thurston's presentation of this matter, but it stands to reason that if any person should have occasion to look up certain information on economy in the operation of steam-engines, and the bearing which feed-water temperatures have in the matter, and should run across this table, without a previous investigation of what precedes the table, he would certainly become very much confused by the data therein given.

Mr. William Kent.—One remarkable thing about this engine seems to be its slow speed. Why was it necessary in designing this engine, supposing it to be the best that has ever been built, to design it for a slower speed than pumping-engines were built for fifteen years ago? The higher speed gives us economy of first cost of engine and foundations, and some other economies, and is supposed to give economy of steam. Why was it necessary to make this engine give a piston speed of only 203 feet per minute?

As to the analysis of this Pocahontas coal, it is given as containing about 7% of volatile matter. That would rank that coal as a moderately soft anthracite. I have seen no analysis of semi-bituminous coal which did not have about twice that quantity of volatile matter. The Pocahontas people themselves have never published, so far as I know, any analysis that looked like that. If the Pocahontas region has any such coal it ought to be known.

Mr. F. H. Ball.—As members of the American Society of Mechanical Engineers we may well be proud of the remarkable

engine which is the subject of this paper, because it is not only an American engine, but was designed by a member of our Society. It might be added that it was tested also by members of our Society, who have given us the result of their investigations in the valuable paper to which we have just listened.

If I may be allowed to criticise so able a paper, I would like to call attention to the manner of combining the indicator cards which is here employed. Referring to page 363, it will be seen that the diagrams are first reduced in length to relative scales that correspond with the relative piston displacement of the three cylinders. Then the diagrams are placed at such distances from the clearance line of the proposed combined diagram as to correctly represent the clearance in each cylinder.

All this is entirely orthodox, and in accordance with accepted methods, but nevertheless is entirely wrong, as I will undertake to show.

Let Fig. 128 represent a set of theoretic diagrams from a triple-expansion engine, in which the clearance of high-pressure cylinder is 2%, the intermediate 20%, and the low 5%.

Combining these cards by the method used by Professor Thurston, we have the combined diagram shown in Fig. 129. Here each diagram is placed in its proper relation to the clearance line, and yet who will say that such a combination can serve any useful purpose?

The same diagrams are combined by another method in Fig. 130. By this method the diagrams are first reduced to their proper relative length, as in the former case. Then, instead of placing them with relation to a common clearance line, so as to represent their respective clearances, they are placed with relation to a common continuous Marriotte compression curve. This compression curve may be arbitrarily located, or may be the extension of the compression curve of any one of the diagrams, having first located the clearance line with relation to the compression curve of this diagram.

In placing the other diagrams against the continuous compression curve of the combined diagram their respective compression curves will not probably coincide with the continuous curve, and the diagrams must be reconstructed in the usual manner, by drawing any convenient number of lines through the diagrams parallel to atmosphere, and then transferring the measurements on these lines to the proposed compression curve of the com-

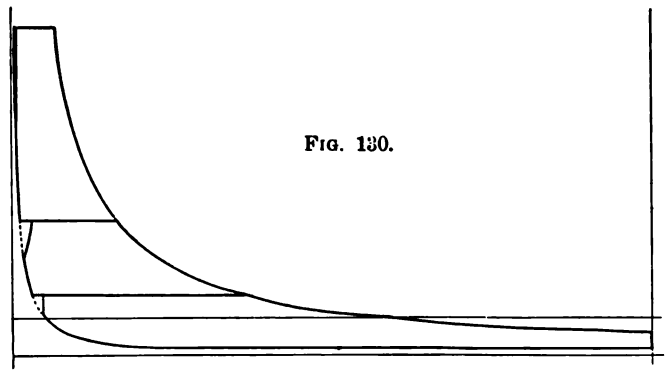


FIG. 130.

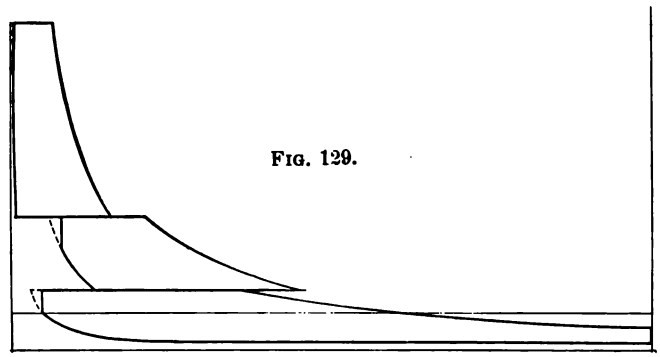


FIG. 129.

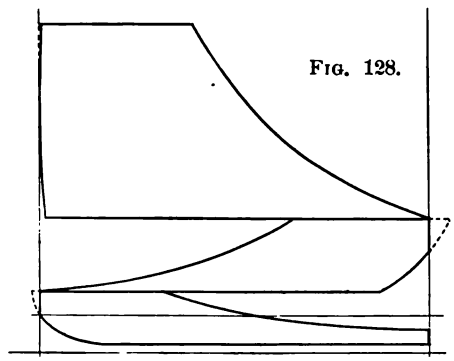


FIG. 128.

Bradley & Prates, Engr's, N.Y.

bined diagram. In doing this the compression curve of each diagram must be completed up to the line of highest pressure in that cylinder.

The supposed object of combining cards from an engine, where the expansion is carried through successive cylinders, is to compare the aggregate performance with the expansion of the same volume of steam in a single cylinder.

Can there be any doubt as between Figs. 129 and 130 which offers the best opportunity for such an investigation?

It may be said that this is an extreme case, but is it not by extreme cases that errors are made clearly apparent?

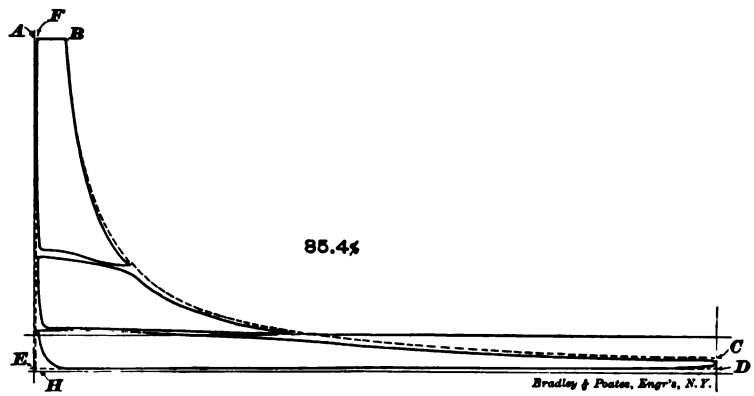


FIG. 131.

With very small clearance the faulty method commonly employed, as in Fig. 129, is less conspicuously wrong, but the error is still there on a reduced scale.

Figs. 131 and 132 represent a set of diagrams from the Milwaukee pumping engine, combined by both the methods that have been described.

The writer is indebted to Prof. R. C. Carpenter for the original diagrams here used, which were taken during the test described in the paper under discussion.

The clearance of this engine is extremely small, and yet an appreciable difference appears in the resulting diagrams represented in Figs. 131 and 132.

These combined diagrams often become the basis of a comparison between the total work realized and that due to the

theoretic diagram, and the result is generally expressed by a ratio. Thus, in Fig. 131:

$$\text{Ratio} = \frac{\text{Aggregate Indicated Work}}{\text{Work represented by area of } A,\ B,\ C,\ D,\ E,\ A,} = 0.95$$

The same diagram, measured in Fig. 132, gives as follows:

$$\text{Ratio} = \frac{\text{Aggregate Indicated Work}}{\text{Work represented by area of } F,\ B,\ C,\ D,\ K,\ F,} = 0.91$$

Fig. 131 is exactly the method used by Robert W. Hunt and

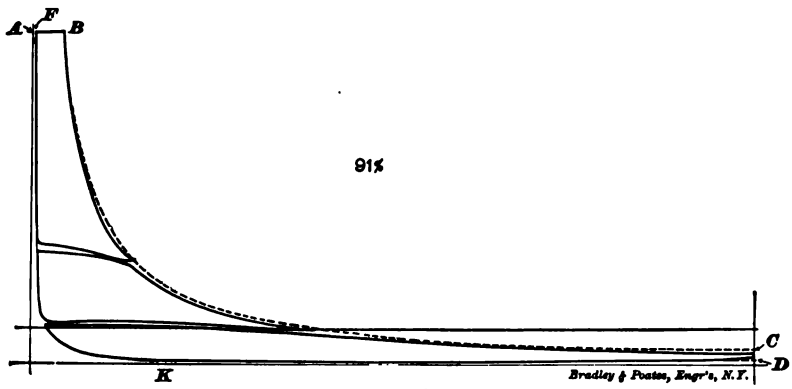


FIG. 132.

James N. Warrington in summing up the result of their test of the pumping engine in the Fourteenth Street station in Chicago. The report referred to appears in a pamphlet published by the builders of the engine, E. P. Allis & Co. of Milwaukee.

A refinement of this method is employed by Professor Denton in his paper on the Laketon engine, presented at the Chicago meeting, and forming part of Vol. XIV. of the *Transactions* of this Society.

Professor Denton's method is to draw the line FH perpendicular to the atmospheric line, and located so that FB shall represent the volume of steam measured at cut-off in high-pressure cylinder, less that which has been restored by compression into clearance space.

He then uses the area of $F, B, C, D, H, F,$ for a divisor instead of A, B, C, D, E, A, as in the former method.

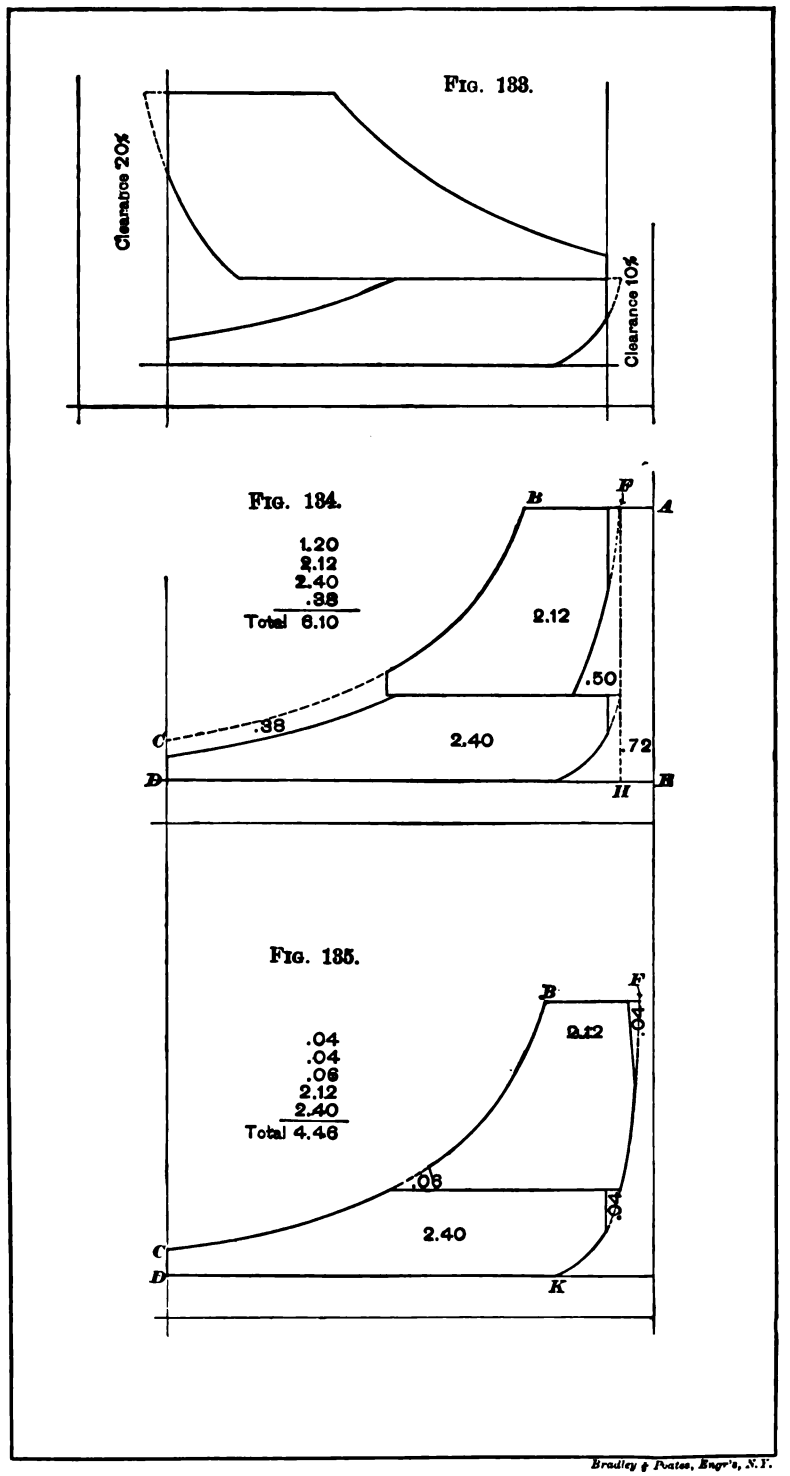


FIG. 133.

FIG. 134.

FIG. 135.

Bradley & Potter, Engr's, N.Y.

This matter may seem foreign to the discussion of Professor Thurston's paper, but is introduced for the sake of completeness, and to show the error that may follow from the use of a clearance line in combining diagrams, instead of a compression curve.

To illustrate this more fully let Fig. 133 represent theoretic diagrams from a compound engine exhausting into atmosphere.

The clearance of high-pressure cylinder is assumed to be 20%, and the low pressure 10%, with a cylinder ratio of 2.

Fig. 134 represents these theoretic diagrams, combined in the manner used by Professor Thurston in the paper under discussion, and also by Professor Denton in his paper referred to, in which the location of both diagrams is determined by the clearance line.

In this combined diagram, $B\ C$ is the theoretic expansion curve, and according to the generally accepted method, the ratio of area realized by this engine is:

$$\text{Ratio} = \frac{\text{Aggregate Indicated Work}}{\text{Work represented by area of } A, B, C, D, E, A,} = 0.74$$

According to Professor Denton's method we have as follows:

$$\text{Ratio} = \frac{\text{Aggregate Indicated Work}}{\text{Work represented by area of } F, B, C, D, H, F,} = 0.84$$

The same diagrams are combined in Fig. 135 in accordance with the compression curve method already explained. The ratio of work realized as determined by this diagram is as follows:

$$\text{Ratio} = \frac{\text{Aggregate Indicated Work}}{\text{Work represented by area of } F, B, C, D, K, F,} = 0.97$$

We have then, from the same theoretic diagrams, the following conflicting ratios of work, as computed by the three methods:

Commonly accepted method	74%
Professor Denton's method	84%
Compression curve method	97%

Inasmuch as these diagrams are theoretic diagrams, the only

loss of area is that due to "drop" at exhaust opening, and to incomplete compression. These losses are represented by the area of the three small triangles, which by careful measurement is found to equal 3% of the area of *F, B, C, D, H, F*, so that of the total area of *F, B, C, D, H, F*, 97% is indicated work and 3% free expansion loss.

If this compression curve method is right, the others are *very* wrong, particularly in extreme cases, and therefore ought to be abandoned.

It is no more trouble to combine the cards on a compression curve than with relation to a clearance line.

Only one of the three methods can be correct. Which is it? Perhaps I misunderstood Professor Denton's method; and I would like to ask him if, in constructing his theoretical expansion curve, he uses the clearance line, or did he use line *F H* as the clearance line for his expansion curve?

Professor Denton.—*F H.*

Mr. Ball.—Then I misunderstood him, and am very glad to be corrected, and am glad to find so good an authority as Professor Denton on my side of the question in regard to comparing areas. It makes just this difference: if the line *F H* is used as the clearance line, the expansion curve would fall below the high-pressure curve, and coincide at the release in low-pressure cylinder.

Professor Denton.—That is what I said. If you draw that line, then the area between that line and the vertical is the same as between your two bands.

Mr. Ball.—Professor Denton says further, and I quite agree with him, that it is not necessary to combine these diagrams at all to make the comparisons of areas. The diagrams can be measured and the high-pressure area reduced by the ratio of cylinders, and added to the area of low-pressure diagram, and then the theoretical area can either be obtained by computation, or by drawing a curve and measuring it. But if the diagrams are to be combined, it seems to me we should use a method of combining them which will fit all cases, and convey some idea to us even in extreme cases. Now, if we can get what we require in the way of comparisons of area without combining diagrams at all, the chief reason then for combining them is to get a graphic idea of what is taking place in the cylinders, and comparing the expansion curves with the continuous

theoretical expansion curve. Take the case shown in Fig. 129 as compared with Fig. 130. As between those two, which is the most satisfactory method? Bear in mind that one method is no more difficult than the other. If a man was piling up cord-wood, and one stick projected beyond the others, he would hardly be satisfied with his work until he had pushed it back into place, and it does not seem to me that an engineer ought to be less particular about his work.

Prof. J. E. Denton.—You would do the engine builder too much justice by showing only the lower diagram. You would not show that he had such an enormous amount of clearance as the upper one showed.

The economy of the Milwaukee pumping engine, as given in this paper, is 11.8 lbs. of steam per hour per indicated horse-power. This is accomplished with 120 lbs. boiler pressure, with a ratio of expansion of about 20, and about 20 revolutions per minute, with cylinders 28, 48, and 74 inches diameter by 5 feet stroke.

The best public record of a triple-expansion engine up to the advent of the Milwaukee engine was that of a Sulzer* mill-engine, which consumed 12.7 lbs. of water per hour per horse-power, with about 150 lbs. boiler pressure, 24 expansions, and 66 revolutions per minute, in cylinders about 20, 30, and 48 inches diameter by 55 inches stroke.

The theoretical laws of thermo-dynamics, supplemented by our best empirical knowledge of cylinder condensation, losses of pressure between cylinders, etc., cannot account for the superior economy of the Milwaukee engine as compared to the Sulzer. The theoretical calculations of the author of the paper, at page 360, regarding the probable water consumption of the Milwaukee engine, omit the theoretical steam consumed in the jackets according to Rankine's formulæ, which amounts to about 1.5 lbs. of feed-water per hour per horse power. But assuming that there was no such numerical error, the method pursued in computing the various losses, if applied to the case of a Sulzer engine, would show that the latter should be more economical than the Milwaukee engine, for the Sulzer engine has a greater ratio of expansion, and the loss by cylinder condensation, as computed by the formulæ of the author, page 359, would be much

* See paper DXXXIX., *Transactions A. S. M. E.*, 1893.

less per horse-power than for the Milwaukee engine. We can get very little light from theory, therefore, to explain the superior performance of the Milwaukee machine.

The percentage of theoretical mean effective pressure for Marriotte expansion realized by the Milwaukee engine is 89, as compared to 85 for the Sulzer. The greater ratio of expansion of the Sulzer, however, makes the water accounted for by the card at cut-off in high cylinder 9.41 lbs. per H. P., or practically the same as for the Milwaukee engine, using the percentage of cut-off given in the paper.

Taking the steam entering the high-pressure cylinder as 100, the distribution of the two engines is as follows:

	Milwaukee.	Sulzer.
Fed to boiler	110.1%	117.5%
Condensed in jackets	10.1%	17.5%
Entering engine	100.0%	100.0%
Accounted for cut-off high cylinder	88.0%	86.0%
Condensed during admission to high cylinder	12.0%	14.0%

It is therefore seen that the remarkable elements in the distribution of steam in the Milwaukee engine are, first, that its cylinder condensation during admission to the high cylinder is almost as little at 20 revolutions as had been accomplished by any other Corliss engine at upwards of 60 revolutions; second, that the jacket consumption is 7.5% less in the Milwaukee than in the Sulzer. The latter difference represents fully the difference between the water consumption of the two engines. The important question, therefore, is, why was the jacket consumption of the Milwaukee engine so much less than that of the Sulzer engine?

Professor Carpenter's test, as published by the builders of the engine, states that the temperature of feed-water of the Milwaukee engine under working conditions was 132°. If this temperature of feed was due to the admixture of the feed-water with the water from the hot well, taking the latter at 105.4°, it would follow that the jackets must have consumed over 12.3% of the feed-water.

Regarding Mr. Ball's discussion of the difference in methods of computing the ratio of the area of the combined diagrams to that of some standard theoretical area, I would say, that his method will give the same results as the one which I have used.

The method used by Messrs. Warrington & Hunt does not take account of steam saved by cushioning, and refers the actual area to a curve whose law is $pv^{1.0}$ = constant, instead of to Marriotte curves. It must necessarily, therefore, give results differing from those found by Mr. Ball's method.

Mr. R. C. Carpenter.—Most of the inquiries raised in this discussion are quite fully answered in the report of the test made to The Edward P. Allis Company, but the following explanation may be of interest in connection with Dr. Thurston's paper:

The official trial of the engine, made July 20, 1892, was of twenty-four hours' duration. It was in charge of Mr. G. H. Benzenberg, C. E., and showed, without any reductions for moisture in steam, 12.15 lbs. of water per I. H. P. per hour. Learning of the results of this test, Dr. Thurston and also the writer desired very much to disprove or verify them, and finally arranged to repeat the test, the results of which have been so fully discussed by Dr. Thurston.

In making this test every conceivable precaution the writer could think of was taken to prevent error, and his closest attention was given to secure accuracy of results. The tanks used for measuring the water were arranged so that the discharge was visible, and by means of a movable trough, so fixed that *had any leakage occurred while a tank was filling, this leak would have gone into Lake Michigan, and not into the boiler feed-water.* Duplicate observers, who kept separate log sheets, were used in every place. The actual time of emptying each tank was recorded in the log, and affords a check by its regularity on error in counting.

I regret very much that the sketch, Fig. 133, has misled Mr. Rockwood in that respect; as this sketch was expected only to show the general arrangement of our testing apparatus and the parts of the plant with respect to each other, there was no attempt at accuracy of detail. The sketch was made by a draughtsman not present during the test, to a small scale, and I had myself not before noticed the error in detail referred to by Mr. Rockwood. His conclusion is absolutely wrong.

Before commencing the test the boilers were carefully examined for leaks, and a small one was found in the blow-off valve. This was stopped by disconnecting the pipe and putting on a cap. Special arrangements were made to run the engine under good conditions; as large a head of water as was possible to

secure was carried throughout the test, and in the opinion of Mr. McMillan, the engineer in charge of the station, things were in rather better condition than at the time of the official test.

The average I.H.P. at the time of the official duty test was 557.029, on official economy trial 576.449, and of our test was 573.87. My test showed 0.35 lbs. of water better results, or 2.88% higher economy, assuming the quality of steam to have been the same in both cases.

During the official trial no calorimeter was used, and no special examination for leaks at the boiler was made. It seems quite probable that nearly all this difference would be accounted for by these circumstances.

THE BOILER TEST.

The firemen employed were the regular men. During the test three firemen were on duty: fireman "A," from 9 A.M. to 2 P.M.; "B," from 2 P.M. to 10 P.M.; "C," from 10 P.M. to 6 A.M.; and "A" from 6 A.M. to 9 A.M. We thus had the same fireman for five hours at the beginning of the test and three hours at the end. A careful accounting of coal burned, water evaporated, condition of fires, and water level in the boilers was made at the end of each watch. The result showed an equivalent evaporation from and at 212° per pound of coal for each fireman as follows:

Fireman "A," beginning of test, 11.05°: end of test, 11.04°.
Fireman "B," 10.3°. Fireman "C," 10.3°. Average, 10.88°.

There is no reasonable doubt but that the record made by fireman "A" could have been maintained throughout the test. A great part of this difference can be accounted for by the difference in temperature of the boiler room. The boiler room is situated on the shore of Lake Michigan, and during the test a strong wind was blowing. Fireman "A" kept the boiler-room closed and the temperature at about 76° Fahr. Firemen "B" and "C" both insisted on keeping windows and doors open, so that the boilers were practically out of doors, the temperature averaging 53°. The temperature of the flue, measured near the chimney, averaged 406° throughout the test. The hot gases were carried away in a sheet-iron flue, 5 feet high and 4 feet wide, containing about 135 square feet of radiating surface per boiler. The total radiating surface

in the flue was about 675 square feet, to the point where the temperature was measured, and in addition the four boiler fronts contained 232, making a total of 907 square feet metallic radiating surface, directly exposed to the air, between the furnaces and point of measuring the temperature in the flue. It may not be of much value, but will no doubt be of some interest, to compute the probable radiation from these two surfaces, as this loss is not included in the measured loss from the flue. From some extended experiments made on the heating of air I have found that ordinarily still air will take from an exposed metallic surface something over 2 B. T. U. per square foot per hour and per degree difference of temperature. This rate is increased rapidly when the air is in motion, practically in proportion to the velocity. It also increases with difference of temperature. It will not be far from the truth to assume that the radiation rate for firemen "B" and "C" was one-half greater than for fireman "A," because of the increase in velocity as well as the lower temperature of the boiler-room. For fireman "A" this loss becomes $2\ (406 - 75)\ 907 = 604{,}062$ B. T. U. per hour. For firemen "B" and "C" the loss is 906,093 B. T. U. per hour. The total heat absorbed by the steam was 7,554,833 B. T. U. per hour, so that the difference in radiation loss is, as figured above, more than sufficient to account for the difference in the results obtained by the two sets of firemen. The flue-gas analysis shows only about 8% of free oxygen in the flue, 6% CO_2, or an excess of air of only .61%. The weight of air in the flue per pound of carbon would be 18.9 lbs. The heat absorbed in B. T. U. per pound of carbon burned would be $18.90\ (350)\ 0.238 = 1{,}570$ B. T. U. If the total heat per pound of combustible be assumed 14,500 B. T. U.,* the loss per cent. of heat, supposing hydro-carbons all consumed, carried off in the flue, is 10.85%. The percentage loss of heat, that is, the ratio of difference between that absorbed by the steam and that which would have been given off on basis of 14,500 B. T. U. for each pound of combustible to the total, was, during the time fireman "A" was on duty, 23.9%, and during the time firemen "B" and "C" were on duty, 27.9%, averaging for the whole time 26.55%.

* This is believed to be a fair estimate, for the trial in the coal calorimeter showed 14,232 B.T.U. per pound of dry coal. This would be equivalent to 14,460 B.T.U. per pound of combustible, since the sample contained 1.47% ash.

So far as we could tell, the additional loss in the flue was essentially constant, and equal to about 10.85%. The radiation loss from metallic surfaces, calculated as above, during time fireman "A" was on duty, was 8.0% of the useful heat, or 6.0% of the total heat. The losses for the other firemen by radiation from metallic surfaces were 12.0% of the useful, or 8.6% of the total heat. The radiation losses calculated by Peclet's coefficients, which are no doubt more accurate than those used, show respectively 9.0% and 12.9%, and give the values in the second columns A_2 and B_2 and C_2. Summing up these calculations we have the following results:

	Fireman		Firemen	
	"A."	A_2.	"B" and "C."	B_2 and C_2.
Heat absorbed by steam, per cent.....	76.1	76.1	72.1	72.1
* Heat carried off in flue, per cent.....	10.8	10.8	10.8	10.8
* Heat radiated from metallic surfaces, per cent.....	6.0	9.0	8.6	12.9
Heat radiated from brickwork and not accounted for, per cent.....	7.1	4.1	8.0	4.2

The figures given in the above calculations, while approximate, show that the total heat in the coal is readily accounted for, and that, considering the conditions, the evaporation was all that under the circumstances could have been expected. Furthermore, the results obtained by fireman "A" were most excellent

In analysis of heat losses in test of Worthington pumping engine at Hampton, England, Prof. W. C. Unwin summarizes the results as follows:

	PER CENT.
Heat due to coal used	100.00
Given to steam	77.00
Carried off in furnace gases.....	10.00
Probable loss due to opening fire doors.....	1.00
Due to carbon in ashes	1.10
Radiation and unaccounted for.....	10.8

My own opinion, founded on the examination of over five hundred trials, is that the results obtained, comparatively speaking, *are very high*. I found on record three boiler trials with the

* Had no heat been radiated from flue, its temperature would have been in one case 530°, and in other case 630°, to carry away the extra heat.

same coal, in which the results obtained in each case were lower; in one test under similar conditions the results were 4½% higher. The results of our test are higher than is obtained in ninety-nine out of a hundred boiler trials. Furthermore, the three firemen employed were wholly unacquainted with the use of bituminous coal, the regular fuel being anthracite, and I am informed that two of the firemen never before had fired with soft coal. Compared with other tests of the same coal they have done well, although in the abstract the results may be low. Considering the fact that we had the regular firemen, who had no reputation of special skill, and were for the first time burning a coal to which they were unaccustomed, it strikes me that the results are as high relatively as the water consumption of the engine. With one exception the results are the best I have yet obtained on a boiler trial. The results show the consumption of 1.25 lbs. of dry coal per I.H.P. per hour, which takes into account the alleged bad performance of the boilers.

Professor Denton thinks that the percentage of jacket-water is unreasonably small. I may state here what I have already fully explained in the report of the test made to The Edward P. Allis Company,* that because of the high temperature of the discharge water from the jackets, some doubt might exist regarding that amount; but this in no way affected the total water consumption of the engine, since the feed-water to the boiler was correctly measured. The discharge from the jackets was led through a long cooling pipe submerged in a tank of water, but even with all the cooling surface possible to secure, the discharge was very hot, and considerable steam escaped. The loss of steam I tried to determine by noting the loss of weight in a given time under similar conditions, but this was far from satisfactory.

As a check on this I have the temperatures of the condensed steam and feed-water, which provide quite accurate data for the computation of the relative proportion of jacket-water to condensed steam. In this engine a surface condenser was used; the discharge from this, mixed with the jacket-water, supplied in ordinary cases the feed-water for the boilers. The discharge from the condenser varied in temperature from 108° to 110°;† the

* See pamphlet published by the company.

† This quantity could not be directly measured, but was computed from the reading of vacuum gauge. The average gauge reading was 13.84, error of gauge 0.52, barometer 14.5, absolute pressure in condenser 1.18 lbs. Corresponding temperature of feed-water 108.4.

temperature of the feed-water varied from 130° to 132°. In other words, the jacket-water was sufficient to raise the temperature of the condensed steam from 108° to 130°.

It seems quite probable that there must have been some loss, as already referred to. This loss was small, and no doubt was less than 1%. To supply this loss, cold water at a temperature of 34° was drawn from Lake Michigan. This loss is considered in the calculation, but it is more than probable that it is offset by a leakage of steam from the jacket traps. For it should be noticed, the escape of any steam whatever from the traps would reduce the amount required to raise a given mass to a given temperature.

The jackets on high-pressure and intermediate cylinders and the two receivers were each supplied with steam at 120 lbs. gauge pressure, and discharged the water of condensation through a common trap at a temperature due to the pressure 350°. Any loss by radiation is assumed to be made up by leakage, which was known to exist. The low-pressure jacket was supplied with steam through a reducing valve, and kept at a constant pressure of 34 lbs. by gauge. The discharge of the condensing water was from a separate trap, at temperature due to pressure. In the computation I have considered that each jacket condensed an equal amount of steam; whether this is the exact fact or not will not materially affect the result. Computation by product of surface and difference of mean temperatures indicates less than 20% for the low cylinder.*

For the computation, let y equal the weight of steam condensed in any one jacket, so that $5\,y$ equals the total weight of jacket steam. Let x equal the weight of steam discharged from the surface condenser.

* If we compute the per cent. of jacket-water as proportional to the product of heating surface exposed and mean difference of temperature, we will obtain the following results:

	Heating Surface. A.	Diff. Mean Temp. B.	Product. A and B.	Per cent. Jacket-Water to each.
	Sq. Ft.			
High pressure cylinders	45	30	1350	1.9
Intermediate	88.7	95	8420	11.9
Low	158	84	13272	18.8
First Receiver	283	70	19800	28.1
Second "	205	135	27680	39.3

Temperature of discharge from four jackets, 350° Fahr.; from low-pressure jacket, 280° Fahr. Temperature of condensed steam, 108° Fahr. Temperature of feed-water, 130° Fahr. Temperature of lake water, 34° Fahr.

As the loss of heat by jacket-water equals the gain to the condensed steam, we have

$$4y\,(350 - 108) + y\,(280 - 108) = (130 - 108)\,x + \frac{x}{100}\,(130 - 34)$$

$$968y + 172y = 22\,x + \frac{96x}{100}$$

$$1140y = 22.96\,x.$$

$$y = .0202x = 2.02\%\,x$$

$$5y = 10.10\%\,x$$

$$5y + x = 110.10\%\,x$$

$$\frac{5y}{5y + x} = \frac{10.10}{110.10} = 9.16\%,$$

which is the per cent. that the jacket-water should bear to the total steam consumed by the engine, calculated by its heating effect on the feed-water.

Professor Denton has computed the results for return-water 105.4, and feed-water 130. The results for that case should be as follows:

$$4y\,(350 - 105.4) + y\,(280 - 105.4) = (130 - 105.4)\,x$$

$$1153y = 24.6\,x$$

$$y = 2.13\%\,x$$

$$5y = 10.65\%\,x$$

$$5y + x = 110.65\%\,x$$

$$\frac{5y}{5y + x} = \frac{10.65}{110.65} = 9.6\%.$$

If the feed water be taken as 132°, the per cent. of jacket-water required would be 9.95. The above results can readily be checked, and it will be seen that Professor Denton is in error. The 105.4 is the result of poor proof-reading on my part, due to the substitution of 5 for 8, but had there been no misprint Professor Denton would not have been warranted in drawing the conclusion stated in his discussion.

The computation made with the actual data is *proof conclusive* that no matter what may be the percentage of the water

condensed in the jackets in other engines, the results obtained by direct measurement vary but little from the truth. I am inclined to believe that, because of slight leakage of steam from the jackets, the quantity given is large rather than small.

Professor Denton referred in his oral discussion to the report, with an insinuation that it is full of errors; this, it seems to me, is far from the truth, although he seems to have found all that were made—and more. In the original report there were two typographical errors, in one an 8 was replaced by a 5, in the other a 7 by a 9, both of which escaped my observation, but were made evident at once on comparing with original copy. Only one of these appears in the paper by Dr. Thurston, viz., the discharge from air-pump temperature, which should read 108.4, instead of 105.4. The measurements from indicator diagram, which Dr. Thurston did not publish, were all given in the original report in absolute terms. The cut-off at bottom end was given as 39.264, instead of 37.264. These in all cases included clearance. The water rates from the diagrams are subject to slight correction, because of this error.

I might say that the indicator diagrams were integrated with a standardized planimeter, by three independent observers, and in case of any disagreement the diagrams were carefully re-integrated, and it does not seem probable that our error in this work should exceed one-half of 1%. All computations were checked, and the only errors yet discovered are typographical, and are fully given above.

I wish to say one word regarding Mr. Ball's method of combining indicator diagrams.

The diagrams shown in the paper were combined under my supervision and as he describes.

In our educational work recently we have been comparing the actual diagrams with a saturation curve that would have been produced by the same weight of expanding steam. The abscissæ of this curve, compared with the actual expansion curve, measured in each case from the clearance line, give us the quality of the steam at any point, and hence this construction becomes of practical value. It would be desirable to have this curve a continuous one, for the case of compound or triple expansion engines. After a great deal of experience I have concluded such a construction is practically impossible. By jogging the clearance lines we can make the ends meet, but the curve is

then mathematically discontinuous; since the weight of steam caught in the clearance spaces affects the expansion curve, and this again is affected by the degree of compression. The method actually used simply reconstructs the diagrams on the same volume and pressure scales, and it makes no hypotheses, but presents the case in accordance with the facts. I do not see why an engine, if constructed with such bad proportions as described by Mr. Ball, should have its diagrams transposed so as to disguise those bad features and appear in as good form as if from a well-designed engine. It appears to me, everything considered, that the old and standard method is better.

*Prof. R. H. Thurston.**—Taking up Professor Peabody's comments, he thinks it would be better not to make our computation of duty on a basis of assumed dry steam. Well, he can take his choice. He will find computations made in the paper on the bases of both dry and wet steam. He thinks it misleading to discuss the Carnot cycle, and remarks upon the obvious fact that steam, working in the Carnot cycle, is not steam drawn from the boiler, pound by pound, but steam in a cylinder, changing its condition from time to time by introduction and rejection of heat. That, of course, is true. The figure stated is the number of pounds of water circulating and of steam generated in the unit of time in such a system. This figure is given in order that we may compare the performance of the actual engine with that of the most perfect possible ideal case, on a common basis, and that basis a usually accepted and well-understood one. The equivalent of such a cycle as we may actually employ, being found in the representative Carnot cycle, the latter would demand the heat-supply in the stated weight of steam generated. If this work were done in the most perfect ideal machine, working in the Carnot cycle, the heat used, instead of being that stored ordinarily in the $9\frac{1}{4}$ lbs. of steam of the ideal case of the Rankine cycle, or the $11\frac{2}{3}$ or 12 lbs. of the actual cycle, would be the equivalent of that contained in about $8\frac{3}{4}$ lbs. of similar steam.

The drawing shown as Professor Peabody's, representing the fluctuation of energy in the engine, is one sent me by the builders of the machine. I was, of course, glad to get all points relating to its performance, and hoped that Professor Peabody would do as he proposes—give us a discussion of that case.

* Author's Closure, under the Rules.

Mr. Ball's debate interests me, but requires little comment. The compound diagram given in the paper is that used in the office of the builders of the engine, and represents the usual method of combining diagrams. My own idea would be to adopt a saturation expansion-curve, of constant weight, for reference, rather than any compression line; in that case the cards would be located, when the data permit, by adjusting the cut-off point of the card by the curve of constant weight, by measuring the volume of steam condensed at cut-off, which volume measures the space between the curve of constant weight and the cut-off point of the card.

Mr. Rockwood is inclined to criticise some points in regard to method of test and other points that have nothing to do with these tests. I will refer him to Professor Carpenter, who answers whatever criticisms are made upon his methods. I think that these results are substantially accurate. In fact, it is easily seen that they are certainly so; as the trial is a repetition of work already satisfactorily duplicated. A preliminary trial was carried through by others, and the final work was in this case performed after a certain amount of drill had been had; so that, having taken all usual and common-sense precautions, such as every one adopts in such cases, and with skilled experts assisting and checking the figures, the results reported must be very exact, and substantially as here given. It is to be observed that similar figures are obtained from other engines, and by other engineers making similar tests of the same style of engine; all giving about 12 lbs. of steam per H.P. per hour.

Some uncertainty is suggested as to the quantity of water from the jackets. I do not think that uncertainty is very great either as to quantity or as to effect, as we will see later.

Mr. Rockwood objects to what he calls *my* theory of the multiple-cylinder engine. I think it is the general theory, the usual theory, of that engine; it being, simply, that the waste of any one cylinder in the series goes to supply the demand for waste in the succeeding one; and it is obvious that, the more cylinders we can introduce, other things equal, the less the waste by initial condensation. The more cylinders introduced, the less the range of temperature in each cylinder, and, therefore, the less this condensation; and if we can get so many in, without otherwise inconveniencing ourselves, as to make that

range insignificant, the waste would also become insignificant; and we could then turn our attention to those other sources of waste—external loss by conduction and radiation, and the friction of the machine, as the only serious wastes left to be overcome.

In reference to the "Hirn analysis," it may be said that the original cards on which it is based differ less than $\frac{1}{10}$ of 1% from the mean of all taken on the trial. I consider them satisfactory representative cards. There still remains, in any such case, some uncertainty as to the quantity of steam—as measured on the indicator cards—contained in either cylinder at the point of cut-off. This uncertainty arises mainly from the fact that no one can say precisely what shall be taken as the point of cut-off for either cylinder. I have had different observers express their judgment as to where the cut-off should be considered as occurring, and find that computations based on their judgment of this matter would give all the way from $10\frac{1}{10}$ and $10\frac{3}{10}$ lbs. of steam up to $10\frac{1}{2}$. The actual figure is probably exceedingly near the figure given in this paper.

It is always difficult to either contradict or to disprove mere suggestions of possible error, and it would be absurd to attempt it; I may, perhaps, suggest that it is absurd to make such suggestions here. The fact is that these trials have been very carefully and skillfully made; the results are confirmed by the work of other investigators; and I have no doubt they will be accepted the world over as exhibiting accurately about the best work that man has yet been able to do with this class of engines.

Some criticism has been made on the introduction into this paper of computations of the economy of ideal types of engine. That is a matter of taste and judgment. My taste and judgment lead me to employ them. What is presented here of that kind is not intended as an attempt at prophecy at all, but we want to know what, in such cases, is the range still left the engineer for further improvement. By making such computations we find in this case that about 25% of the heat is wasted, even in the best of our engines, and that the engineer still has an opportunity to improve this best of contemporary machines to that extent.

The computation of the representative ideal case is fully explained in the paper. The Rankine cycle is accepted for the jacketed engine for the purposes of these computations, which means that our computation is made on the assumption that, in

such an engine, the ideal jacket might be expected to supply the heat required to keep the working steam in the dry and saturated state. But it is also recognized that, when we have actually built such an engine as this, the heat from the jacket may be expected to be insufficient to do all the work, and that, in every engine actually built or likely to be built, there still will be found some additional waste. As I said in introducing the paper, it was assumed that those wastes should be taken to represent minimum quantities, each in its kind. These quantities were added to the original computation of wastes and steam consumed, and the total is given as representing, thus computed, a fair figure, representative of best modern work.

"Summing up the case," after reading carefully the revised discussion and new matter coming in after adjournment of the session, I have a word of comment to offer in relation to some of the points discussed, and a few new facts to present in confirmation of the deductions made in the body of the paper. Taking the criticisms in order of presentation:

Professor Peabody's suggestion, making the computation of efficiency measured in steam consumed on the basis of wet steam rather than dry, would only modify the results in this case by about one-tenth of a pound per hour per I.H.P., but I prefer the definite basis of dry steam consumption to the indefinite basis of wet steam. If we are to compute duty on a basis of wet steam used, we must still have an exactly defined and conventional basis of comparison. The question would then be: how wet should we take steam to be when establishing our fundamental unit of measurement? I should prefer to take the wetness as *zero*. This gives as definite a basis of comparison as the universally accepted computation of evaporation, "from and at the boiling point under atmospheric pressure." I accept no variable standards that I can conveniently avoid. The absolute standard of efficiency is the heat-supply in thermal units. This is given in the paper; although the weights of steam and of fuel are also given, for the benefit of those who prefer them. I have no inclination to force my ideas of standards upon my friends or the readers of my papers, and see no objection to giving them their units as well as my own. The reason for stating similarly the efficiency of the Carnot system is already given. The comparison of equivalents in the two cycles studied enables us to obtain an intelligent idea of the relative efficiencies and wastes

of the cycles in the ideal case, and to compare both with the real engine performance, in terms commonly used and well understood by the average reader and practitioner. The question regarding the nature of the cycle is answered already, both in the body of the paper and in my discussion above.

My measure of the efficiency of the engine is precisely his, the ratio of the thermal equivalent of work performed to the quantity of heat demanded of the boiler for its performance. It is here so taken. The correct figure is 0.194 for the trial, and the proofs are so corrected. The figure for the engine as customarily operated, with feed 33° Fahr. warmer than at the trial, is 211 B. T. U., instead of 217.6, and efficiency $\frac{42.42}{211} = 0.20$.

The remark about the diagram of stored energy simply repeats the text of the paper directly opposite: "It should, however, be remembered that our formulas are only known to be sensibly accurate for a comparatively low range of pressure and temperature, and the curves at hitherto unattained points may take quite different paths."

Mr. Rockwood assumes that I have endeavored to make the paper presented by me "proof of the correctness of his [my] previously published 'Philosophy of the Multiple-Cylinder Engine.'" * He is entirely mistaken. I had no suspicion of the necessity of any such vindication. His criticisms of the trial, as shown by Professor Carpenter, as well as by the paper itself, are also mainly based upon "the unscientific use of the imagination." The engine reported upon does not "exceed in economy by at least 10% all other engines of its type." At least a half-dozen trials of several engines of this type, built by the same makers, have approximated 12 lbs. steam consumption. The paper gives the figures for earlier trials of this engine and later trials of the Chicago engines, which are older and of a less perfect construction. This engine happens to excel in a reduced clearance and other details, and to have more effective jackets. Whatever might be "expected," the facts are shown by the trials. Mr. Rockwood's computations (3) seem to me incomprehensible and absurd. If correctly made, they might result in a balanced account, giving zero in place of 0.148. They prove neither the point attempted to be made by him nor anything else.

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* *Trans. A. S. M. E.*, Vol. XI.; No. CCCLXII.

His records of his engine trial would be more interesting and valuable if they gave us an accurate measure of the efficiency of the engine. Complicated as they are, with boiler efficiency, his figures for economy are no gauge of the efficiency of the engine. His machines evidently do good work; but I think his idea of substitution of a steam-chamber for the intermediate working cylinder is not to be taken as explaining that good work. His theory is entirely false, in my opinion. The only effect of his intermediate reservoir—and such reservoirs are used now, in all engines of the highest class, between each pair of cylinders in series, whether two or three—is to secure some drying of moist steam, and the effective drainage of superfluous water. It has no thermo-dynamic value. His work is good; but I consider his theories mere theories, and with no basis in fact or reasoning.

In *one* point he is perfectly right. The atmosphere of Milwaukee *is* peculiar; it is an atmosphere of genius; and we should all be glad, I am sure, to have just as large an admixture of that atmosphere here as possible. It should be appreciated here, if anywhere.

Mr. Scheffler has evidently not read the paper which he attempts to criticise. Had he read page 348, as printed, he would have found a paragraph reading thus:

"It will be noted that heating the feed-water entering the boiler, decreasing the range of temperature between that of water supply and that of steam delivery, decreases the maximum thermo-dynamic efficiency of the corresponding Carnot cycle, reduces the quantity of heat stored in the unit mass, and increases both the weight of steam storing unity of heat and the quantity of steam of which the energy thus stored corresponds to the measure of a horse-power. The actual gain by heating feed-water comes of the defect in the operation of the steam-engine cycle, resulting in failure to raise the fluid from minimum to maximum temperature by compression in the working cylinder, and compelling the supply, at the boiler, of heat equivalent to this waste, together with that additional quantity needed to raise the water from the temperature of its source to the minimum limit of the engine cycle. Where this can be obtained from otherwise wasted heat, the saving may prove very important. In this case, the lower limit is fixed by extra thermo-dynamic conditions, and, at the boiler, is always below that

at the engine cylinder, and its variations determine corresponding variations of the quantity and cost of the heat supply."

The most elementary knowledge of thermo-dynamics teaches that, as explained in the paper with ample minuteness, the narrower the range of temperature in the Carnot cycle the lower its efficiency, and the larger the weight of steam required to store a given amount of energy. It *would* be advantageous, if it were practicable, to expand down to a minimum "at 32° or even to zero," or to zero absolute. He is probably misled by taking the term "feed-water temperature," not as intended, as the temperature of the feed of the parallel common cycle, but as that of a Carnot cycle, of which it simply represents the lower limit of temperature. In the Carnot cycle, the water is always brought, by mechanical compression, fully up to the final temperature of the steam, and only the latent heat of vaporization is demanded in "making steam."

Mr. Scheffler finds no hint of this feature of the paper in its title, "which is very exact in its meaning" as he says. He failed to read the title which he commends. It is "On the Maximum Contemporary Economy of the High-Pressure Multiple Expansion Engine; with a Comparison of its Efficiency with that of its Ideal Representative under Similar External Conditions." Its principal purpose, in fact, was to exhibit the last-described comparison, and to show, as well as might be, the range left for the further improvement of the best of existing steam-engines, and the nature and magnitude of the wastes still to be extinguished by the inventor and the engineer.

Mr. Kent is probably right about the volatile matter in the coal, as found by analysis; and I have substituted in the proof the figures for common varieties of that coal, as previously reported. As to engine speed, that is a matter for the designer to explain; but I presume he desired to secure perfect action of pumps, as this is more important than higher engine speed, where the machine is well jacketed. However, the designer is the most thoroughly posted man in the country, if we judge him by his work, and no doubt knew precisely what he was about in settling that matter. The fact obviously is that his engine gives higher economy at its "slow speed" than do the best of other engines at their higher speeds. But it had not occurred to me that it has a low speed for a pumping engine. Many are slower.

Professor Denton finds no way of accounting for the superior efficiency of this engine. This is to be regretted; but, as we have the facts before us, both as to this and to other engines of very nearly equal economy, that is a matter of less consequence. It is, however, in my opinion, very easy to see the main reasons of its remarkable performance. It is the latest and most refined of the designs of one of the most skillful designers in the world; its clearances are reduced to a small fraction of those usually found in such engines; special attention is given to securing low friction and perfect tightness of piston, valves, and, especially, against air-leaks into the condenser; the engines are under peculiarly good management; and every known refinement in design, construction, and management conspires to make its work exceptional. Probably the small clearance and port-spaces are a principal element.

This critic is entirely mistaken in supposing that, as he says, "the theoretical steam consumed in the jackets" is omitted in my computation. The computation is made by calculating, by the Rankine method, the work done in the engine, and the total heat demanded to supply the required steam from the boiler, and to transfer heat to it through the jackets while expanding; the ratio of the former to the latter quantity, whether taken for unit weight of total or of working steam, being the measure of the "efficiency." The quantity of steam required per horse-power per hour, for efficiency unity, is then computed, and the quotient of *this* figure by the above computed actual efficiency gives the total steam of *both* working charge and jacket; assuming that the jacket-water drains into the feed, and that it costs as much to make jacket steam as to make that supplied to the cylinder directly. The difference between this total quantity of steam and that computed by the method of Rankine, which measures the steam entering the cylinder only, is about 10% of the former.

The doubt expressed as to the reported measure of the jacket-water is shown by Professor Carpenter, and ample expert testimony, to be without the slightest foundation. No one can, I think, reasonably doubt the substantial accuracy of the reported figures. The superior action of the jacket is easily accounted for, as just stated, by the comparatively small areas of those surfaces of clearance spaces upon which the major part of the cylinder condensation occurs, reducing correspondingly the

work required of the jackets; by the completeness of jacketing of heads as well as sides—the former being well known to be vastly more important than the latter, though so generally neglected; and by inherent jacket efficiency.

Professor Carpenter's discussion will probably relieve the doubts of any members who may still feel anxious lest some of the erroneous assumptions which have been presented may have a basis of fact. "To make assurance doubly sure," I append the certificates and written comments of those who were officially representative at these trials, of the city of Milwaukee, of the builders of the engine, and of the firm of R. W. Hunt & Co., consulting engineers making a specialty, in one department, of this class of work in investigation.

This matter is of such importance to the engineer and to the country, and has attracted so much attention at home and abroad, that it is possible that a few additional facts bearing upon it may prove of sufficient interest to justify their presentation for this purpose, as well as in reply to correspondents asking further information.

The question has arisen whether it is probable that a jacket can save much more steam than it itself uses. Professor Ewing, in his grand contribution to the *Encyclopædia Britannica* on the steam-engine, says: "When a jacket is working properly it uses in a single cylinder 4% or 5%, and in a compound engine 8% to 12%, of the whole steam supply." Compiling the results of experiments reported to the B.I.M.E., wherever containing the needed data, I find that *the average of sixty cases*, with single engines, gives 4% or 5% of steam used in the jackets, with the result of saving net about 15%. The compound engines, similarly, average an expenditure of 10% in the jackets, saving two and a half times that amount at the boiler. Some of Mair's experiments show a saving of four to five times the jacket steam at ratios of expansion ranging from three to six. Cases in the list just quoted give ratios of six, seven, and eight to one, and none justify the assertion made by some one that it is questionable whether the jacket can save much more than it uses, in any case. The ratio is less as the engine is more complex, compounds giving a smaller ratio than simple, and triple engines lower than compounds, the opportunity for saving being less; but all, in the average use of the jacket, give handsome returns, and a saving which is a multiple of the expenditure in the jacket. Professor

Alden's little compound expends 6% in the jacket and saves 15% at 290 revolutions per minute.

It has been suggested that the reported 9.25% jacket condensation of the Milwaukee engine is probably an error, on the basis of the asserted fact that it is about one-half that of other engines of similar class. This conclusion is quite erroneous. This engine uses about the same jacket steam as other engines of the same style and size, for which about 10% is usual. To base a charge of error on this figure would be vastly less reasonable than to say, for instance, that the Pawtucket engine, a compound, presumably like others effectively jacketed, should use 10%, while reported as the result of measurement to use but 5%. The jackets may be comparatively useless, but there is no reason for suspecting error of measurement. Leavitt & Henthorn's test of the Narragansett Electric Lighting Company's engine gives 14.33%, including 7.4% of its own weight of water, a net 13%. The Plymouth Cordage Company's engine gave 10.24%; the Namquit Mills engine takes 11% into its jackets; the Laketon engine uses 15%; Mr. Leavitt's compounds, "Frontenac" and "Superior," are reported to me by their designer as using 7% when demanding 14.4 lbs. of steam per horse-power per hour under a wastefully light load. The Gaskill reports, as published, give: at Philadelphia, 9.15%; at Chestnut Hill, 13% and 16%; at Lake, 10.32%; at Schenectady, 5.2%; and at Kalamazoo, 11.36%, in trials made by Messrs. Codman, Barrus, Williams, Gifford, and Cooley, respectively. Mr. Bryan reports by letter for the new triple-expansion engine of the East London Water Works at Lea Bridge, 12.28% used in jackets, with a consumption of 13.28 lbs. of steam. Buel's report on the Lawrence engine gave 9% jacket steam, and many other cases might be quoted. The fact is, that a low figure may indicate either, as probably in this case, high efficiency of jacket on reduced areas of condensing cylinder-wall, or low efficiency of jacket—or even, as in the Isherwood trial of the Brooklyn engine, or in that of the Pawtucket engine, comparatively useless jacketing. The sixty cases above mentioned include all the way from below 1% to above 20% of steam used in jackets, the proportion varying with every possible variation of conditions of operation, or of design of engine and of jacket I have even known a superficial coating of grease, too thin to be measurable, to intercept 10% of the flow of heat across the surface of the metal. I imagine that the condition of the surfaces

of the cylinder-wall under the jacket, external and internal, may often determine either a high efficiency, or total inefficiency, of the jacket.

It is especially interesting and convincing to note that when the three trials of this engine, and the trials of the three similar Chicago engines, are collated, the figure for steam per horse-power per hour being corrected, in each, for moisture in the steam, they all give very nearly the same remarkable figure—less than 12 lbs. Our figure for the Sibley College trial of the Milwaukee engine is 11.678; that for the check-test is the same, within limits of minutest error; that for the contractor's trial of the same engine is 11.91; that for the Chicago engines—*for the three together*—is 11.94. The last trials were of forty-eight hours', the others of twenty-four hours', duration. The results coincide with practical identity. The differences are well within usual limits of error of the most skillful observation and computation.

The appended letters from the officers of the Water-supply Department of the city of Milwaukee may aid in removing any doubt, should any still stand, upon the guesses and hypotheses of error constituting the "criticisms" offered. As Mr. Benzenberg says: "There can be no question as to the accuracy and care taken in getting at all the true data and facts" during that trial. Chief Engineer McMillan certifies that he was in the engine-room throughout the whole twenty-four hours of the trial, and that, as to a possibility of error of water-measurement, imaginatively described in these "criticisms," "to any one familiar with the situation and the care taken, an error in that direction hardly appears possible." The water measured was that fed to the boilers, and no other. And no water could get to the boilers unmeasured. Mr. Lewis, of the Allis Company, certifies the same fact; as do Messrs. Printz and Cox, their official representatives as observers. The representative of R. W. Hunt & Co. asserts the results to be accurate within a fraction of 1%; they are "as near correct as is possible." Mr. Edwin Reynolds's letter for the Allis Company, a statement confirmed by Mr. Irving Reynolds to the writer, states the case sharply and justly. The action of the firm in reference to the St. Louis contract is, perhaps, to many minds, likely to be the most convincing evidence of the correctness of statement in the body of the paper.

This discussion has, in the main, been based upon "side-

issues," and upon hypothetical defects of trial or report; but it is, after all, a pleasure to see, as a result, some valuable additional facts brought out; even though the interest attaching to debate may, in some respects, be psychological rather than professional. I confess to a little disappointment in finding so little direct contributive material presented by those who are experienced in the design, the construction, or the operation of this class of machinery; although private correspondence has brought much information of real value to me individually. I hope that we may secure much valuable matter from them, later.

These side-issues tend to divert attention from the principal, the essentially important, points brought up by this peculiarly remarkable success in the perfecting of the modern steam-engine; which advance is a matter of vital interest to the Society, to the profession, to the country, and to the whole human race. We should, I think, after the completion of the ungrateful task of fault-seeking, finally congratulate ourselves that we have among us so talented a designer, such admirable builders, and such enterprising buyers of these high-class engines, and that we are thus enabled to secure such approximation as is here indicated of the real to the ideal thermo-dynamic engine. I am confident that every one will desire to extend congratulations to these colleagues, who have done so much and such good work and who have "set the stakes" so far ahead, as an example and a challenge to the whole world of engineering.

The stimulus thus given to competition among the greatest designers and builders must inevitably result in further advancement throughout the world. We will indulge the hope that our colleagues and our country may retain the lead which they now unquestionably hold, and that, hereafter, we may receive from them many still better reports of still better work done in approximating the efficiency of the real engine to that "of the ideal representative under similar external conditions," as well as of success in raising the figure for the representative ideal case.

APPENDIX E.

MILWAUKEE, *January* 17, 1894.

PROF. R. H. THURSTON, Cornell University, Ithaca, N. Y.

Dear Sir :—In reply to your favor of the 11th instant, I herewith inclose you a statement from Mr. Thomas McMillan, chief engineer in charge of the pumping stations, with reference to the accuracy of the method employed in measuring the water fed to the boilers during the trial of the 18,000,000-gallons pumping engine at our North Point Pumping Station.

There can be no question as to the accuracy and care taken in getting at all the true data and facts during that test run. The engine and boilers were run by our own men, and the utmost pains were taken by all observers to note the performance of the different parts of the engine and boilers with the greatest possible accuracy.

Should you wish any further corroboration of the statements, please let me know, and I will try and obtain it for you.

Yours truly,

G. H. BENZENBERG,
City Engineer.

NORTH POINT PUMPING STATION, *January* 15, 1894.

MR. G. H. BENZENBERG, *City Engineer.*

Dear Sir :—I have carefully read Professor Thurston's letter as to the possibility of an error in the records of water evaporated during the test referred to. To any one familiar with the situation and the care taken, an error in that direction hardly appears possible. During the test, engines No. 2 and No. 4 were running. No. 2 engine was supplied with steam from the south boilers, at 50 lbs. pressure, while No. 4, the engine which was being tested, was supplied with steam from the north boilers at 125 lbs. pressure, so that it was impossible that water could flow from below into the higher pressure. In ordinary running the suction pipe of the feed-pumps is connected to the hot well, but during the test this pipe was connected directly with two barrels which were connected with each other, and placed at a lower level than the measuring tanks. All the water measured passed directly into these barrels, and no more water could get into them than that which passed through the measuring tanks. At the measuring tanks two observers were stationed, one representing the E. P. Allis Company, and the other being one of the gentlemen from Cornell University, with one of our men to operate the valves for filling and emptying the barrels. Each of these kept his own independent record of the water supplied to the boilers, so that an error of the log could be immediately detected. Professor Carpenter clearly explains the position of the tanks in his report. Upon a very careful examination a small leak was found in the blow-off pipe, previous to the test, but this pipe was disconnected and plugged. I re-

mained at the engine-room watching all the observers and their methods of keeping the records during the entire twenty-four hours of the test, and I have no reason whatever to doubt the accuracy of the logs kept.

Respectfully submitted,

THOMAS MCMILLAN,
Chief Engineer Pumping Stations.

January 15, 1894.

Having represented The Edward P. Allis Company at Professor Carpenter's test of the Milwaukee pumping engine, March 25 and 26, 1893, I hereby certify that the feed-pump which supplied the boilers during this test had no suction connection other than that attached to the barrels in which the feed-water was measured, and that there was no steam or water pressure about the plant higher than 65 lbs., excepting on boilers supplying steam for the test, consequently there could have been no water supplied to the said boilers by leakage or otherwise beyond the amount measured in the barrels attached to the feed-pump.

JNO. H. LEWIS.

January 15, 1894.

We, the undersigned, having been present during Professor Carpenter's test of the Milwaukee pumping engine, do hereby certify that we kept tally sheets independent of those kept by Professor Carpenter's assistants, showing the amount of water fed to the boilers, and the weight of coal burned, and that the figures for water and coal consumption given in Professor Carpenter's report of the test are in accordance with our tally sheets, and are, to the best of our knowledge and belief, correct.

CARL J. PRINTZ,
JAMES L. COX.

CHICAGO, ILL., *January* 17, 1894.

DR. R. H. THURSTON, Sibley College,
Cornell University, Ithaca, N. Y.

Dear Sir:—Yours of January 11 is at hand, and some points in it make it seem desirable that we should go to some length to emphasize certain points regarding the test of the Milwaukee engine. It would seem that there is quite a feeling that the figures shown by Professor Carpenter's test are not correct, but why the figures should be questioned by any fair-minded engineer who has read the report carefully I am unable to understand. Of course, it is easy to say "I don't believe it," but if one has read the details of the report carefully, he could hardly make the statement without questioning the honesty of those making the test, and this is something no one would be warranted in doing without having some proof or reasonable argument in support of his contention.

In the first place, the official test of the engine had taken place several months before, so that, as far as The Edward P. Allis Company was concerned, they had nothing at stake, and no hope of compensation for any higher duty shown.

As explained in Professor Carpenter's report, the test was made partially to afford the students practical work in testing a high-class engine, and certainly there could have been no incentive but to show the facts in the case. At the

time the test was made the engine was running at its usual speed, and under every-day working conditions, and no special preparations were made for the test, excepting to take such precautions as to insure absolute accuracy in the results obtained. The engine and boilers were operated by the regular crews employed by the city, and were under the direction of Mr. Thomas McMillan, chief engineer in charge of all the city pumping stations, and Mr. Hamilton, chief engineer in charge of the North Point Station. The Edward P. Allis Company furnished a set of observers to keep duplicate tally sheets of water fed to boilers, coal burned, gauge readings, etc., simply to insure the correctness of the records by comparing them with those of the Cornell observers.

As to the amount of water fed to the boilers, and the evaporation, we would say that during the test the boilers were fed by a pump driven from one of the main pump plungers, which feed-pump ordinarily takes its supply from the hot well on the overflow pipe of the air-pump; but during this test the suction pipe from the hot well to the feed-pump was *disconnected*, and the *only suction connection* for the feed-pump *was that attached to the supply barrels*, the water in these all having to pass through the measuring barrels.

Now, as to leakage, at the time of the test one of the old compound engines was in operation, taking its steam from the old boilers, located under the boiler house at the opposite end of the building, the pressure on these boilers being about 60 lbs. per square inch; and as this was the highest steam pressure about the building, excepting that on the boilers under test (125 lbs.), there could have been no steam leak *into* the steam-pipe or boilers of the engine under test. The highest water pressure about the building was that on the city mains (about 65 lbs.), so that *no water* could have leaked *into* the boilers from any source.

To condense the above: There was no steam or water pressure about the plant equal to more than one-half the pressure on the boilers under test, consequently there could have been no leakage *into* the system under test.

Regarding the evaporation, your mention of the Boston correspondent's doubt leads us to infer that the amount of water evaporated per pound of coal is considered too small, or, in other words, the amount of coal burned would indicate that more water *might* have been evaporated than was shown by the records. As Professor Carpenter's test was to be an efficiency test of the engine, only, no special effort was made to obtain the highest possible boiler efficiency, and the amount of coal burned per square foot of grate (about 7 lbs.), and the water evaporated per square foot of heating surface (1½ lbs.), show that more boilers were in use than were required, and it is a fact that these four boilers burn at least 50% more coal and generate 50% more steam when required to operate a 12,000,000-gallon compound engine in addition to supplying steam to the 18,000,000-gallon triple. With four boilers supplying steam only to the triple engine, the grate surface should have been reduced in order to obtain the best results from the coal, but owing to the fact that these boilers have to do considerably more work at times, it is desirable that the grate area remain as it is. The evaporation shown, however (10.88% from and at 212° per pound of combustible), is not far from the average evaporation shown in Mr. George H. Barrus's boiler tests for plain horizontal tubular boilers with Cumberland bituminous coal. The evaporation, however, would have been 11.41% from and at 212° per pound of combustible, had firemen B and C (who were not familiar with the coal) been as skillful as fireman A; and as the duty per 1,000 lbs. of feed-water was the same during the watches of all three firemen, it is evident that the greater amount of

coal burned during B and C's watches was entirely due to their handling of the fires. The boilers are of the ordinary horizontal tubular type, with common brick settings, and no feed-water heater or economizer is in use; and with boilers of this type we find it difficult to get an evaporation of more than 9 lbs. of water per pound of coal under actual every-day conditions, and cannot under any conditions approach the high evaporation claimed for certain types of internally fired boilers, which boilers your Boston critic may have in mind when he questions the evaporation at Milwaukee. As confirming Milwaukee test, the test of the three similar engines in Chicago may be taken, and as this test was by another party of experts, who were employed by the city of Chicago to determine only whether the contract duty had been attained, we see no reason why their figures should be questioned. The duty per 1,000 lbs. of feed-water was 148,958,000 foot-lbs., and the duty per 1,000 lbs. of dry steam was above 151,000,000 foot-lbs., the figures being the average for three engines during a continuous 48-hour test. The dry steam per I. H. P. was 11.97 lbs., being an average for the three engines. The boilers at Chicago are also of the ordinary horizontal tubular type, but are fitted with down-draught smoke-consuming furnaces, and the evaporation per pound of combustible, with feed-water 212°, was 9.9 lbs. against 10.76 lbs. at Milwaukee. As far as the steam per I. H. P. is concerned, there are many reasons why the Chicago performance should not equal that of the Milwaukee engine.

First. The Chicago engines were designed more than two years before the Milwaukee engine, and did not represent our latest practice.

Second. The clearance in the low-pressure cylinder at Chicago is 2.51%, against ¾ *of one per cent.* clearance in the low-pressure cylinder of the Milwaukee engine.

Third. The piston speed of the Chicago engines is 162 feet per minute, against 203 feet piston speed with the Milwaukee engine.

Fourth. The Chicago engines are supplied with steam at 120 lbs. per square inch, while the Milwaukee engine is supplied with steam at 125 lbs. per square inch.

While some of these differences have, perhaps, but little influence on the results, they all point in the direction of better economy with the Milwaukee engine.

As far as I can see, the methods employed in determining the amount of steam used, etc., at Milwaukee were of the simplest and most direct kind, and as little liable to error as any system that could have been used.

That we have entire faith in the performance of this type of engine is shown by the fact that we have recently contracted to furnish two engines, in general design like the Milwaukee machine, for the city of St. Louis, Mo., at a very low price ($23,000 below the lowest other bidder), relying on the bonus offered ($1,000 per 1,000,000 for all duty above 125,000,000) to bring the price of the engines up to the figures at which they would ordinarily be sold, and we must get at least 140,000,000 duty in order to do this, and we fully expect to get 150,000,000 duty on the test.

Simply because the record is higher than has heretofore been shown, should not be considered reason for discrediting the figures, and perhaps, after Mr. Leavitt's new engine, now building for the Chestnut Hill Station, Boston, has made the record that is expected of it and that it should make, less doubt will be expressed regarding the developing of the I. H. P. for 11.68 lbs. of dry steam.

The Milwaukee engine ran, during the year 1893, 8,400 hours at full speed, or an average of 23 hours per day for 365 days, which would seem to show that high duty can be secured without sacrificing reliability. We should be glad to see the proof sheets of your amended paper, before it finally goes to press, if you can conveniently send them to us, and in the meantime should be glad to furnish you any information at our command.

Yours truly,

THE EDWARD P. ALLIS COMPANY,
By Edwin Reynolds, *Supt.*

THE ROBERT W. HUNT & CO. BUREAU OF INSPECTION AND TESTS.

CHICAGO, ILL., *January* 26, 1894.

PROF. R. H. THURSTON, Ithaca, N. Y.

My dear Sir:—In reply to your inquiry of January 11, concerning the accuracy of the "water consumption" during the test of the Milwaukee pumping engine, we would say that our Mr. McMynn was present at the trial, from 6 o'clock, P.M., March 25, 1893, until the end of the trial. He says that as far as he saw everything was accurate, and the method of testing was the most approved. As far as he could see all the readings were checked, and he says that he has no doubt that the results are correct within 1%. Of course you will understand that we cannot furnish a statement certifying to the absolute accuracy of the test, since our representative was present only at the invitation of Professor Carpenter; but we have no doubt in saying that the results are as near correct as possible, since they correspond closely with tests which we have made on a similar engine.

I remain, yours truly,

ROBERT W. HUNT.

FIG. 108.—ENGINE HOUSE, MILWAUKEE. NORTH POINT STATION.

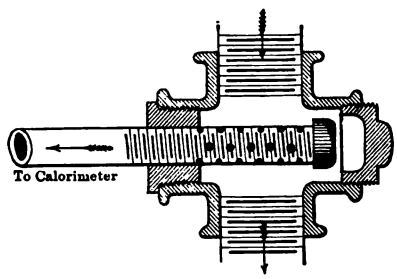
DLXXIII.*

EXPERIMENTAL DETERMINATION OF THE EFFECT OF WATER IN STEAM ON THE ECONOMY OF THE STEAM ENGINE.

BY R. C. CARPENTER, ITHACA, N. Y.
(Member of the Society),

AND L. S. MARKS
(Fellow of Sibley College).

THE following experiment was performed by L. S. Marks, B.S., graduate of Mason's College, England, a graduate student in Sibley College, and by H. H. Tracy, M.E., graduate of Cornell University. The experiment was undertaken for the purpose of finding what effect on the economy of a plain slide valve engine was produced by introducing, by condensation, large quantities of water in the steam supplied the engine. This was accomplished by surrounding that portion of the steam pipe which had a vertical direction with a jacket or vessel of galvanized iron, with a radius about $1\frac{1}{2}$ inches greater than the steam pipe. This vessel was filled a greater or less distance with cold water, and acted very efficiently to condense the steam in the main pipe. The amount of steam condensed was practically proportional to the height of water maintained in this jacket, which was readily regulated by valves on the water pipes supplying water, and on the pipes leading from the jacket for discharge of water.



COLLECTING NIPPLE FOR CALORIMETER
FIG. 109.

No difficulty was experienced in condensing by this method

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

any desired proportion of the steam, but considerable trouble was caused when the problem of securing an average sample for determination of quality in a calorimeter came up for solu-

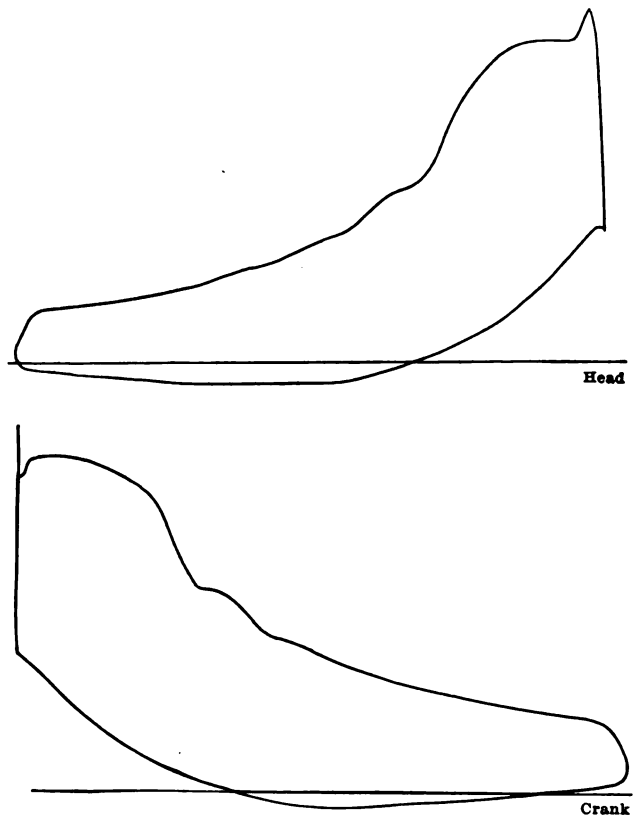


FIG. 110.—Sample diagrams 40 Spring.

tion. After various trials, which consumed much time and patience, a collecting nipple perforated with large holes was screwed completely across the main steam-pipe, as shown in Fig. 109, somewhat beyond the water jacket. The collecting nipple was horizontal, the main pipe vertical. A separating calorimeter was used for determining the per cent. of moisture, and gave results when in this position nearly proportional to the height of water in the steam jacket, and, from some check experiments made by comparison with the heat given to the

jacket water, is believed to give results which are essentially accurate.

The engine tested was a plain slide valve engine with throttling governor, and fitted with an Alden brake for absorbing the power. The dimensions were: Diameter, 6 inches; length of stroke, 8 inches; clearance by measurement at head end, 13.05%, at crank end, 12.51%.

The brake load was maintained at such a point that the governor was well open and had little or no throttling effect on the steam. Sample cards taken with a 40 spring are shown in Fig. 110, which were nearly constant in all the runs.

While the experiment consumed a great amount of time, it is not considered necessary to take a great amount of space in describing the details, save that we may mention that great care was taken to secure accuracy in all the results.

The scheme of the experiment was in each case to maintain all conditions, as nearly as possible, constant, except that relating to the quality of the steam.

The observations made were of engine speed, temperature of exhaust and condensed steam, of injection and discharge water, pressure of steam in steam chest and condenser, weight of condensed steam, and volume of condensing water.

No observations were made until all the conditions became constant, in any case.

The following are the results of the test in tabular form:

LOG OF TEST A. 27TH MARCH.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches high.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total mean H.P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	Mean H.P.	M. E. P.	Mean H.P.	
1	220	75	24.6	162	95	36	89.5	325 lbs.	34	4.24	36	4.11	8.35
2	220	75	....	162	98.5	36	90.5		33.4		36.5		
3	220	75	24.7	162	106	36.8	92.0		33.5		35.2		
4	214	75	24.5	162	102	36.5	91.3		....		35.7		
5	218	76	22.7	162	103.5	36.3	93.0		33.4		34.8		
6	215	76	23.3	162	103.8	36.3	94.3		33.3		35.4		
7	216	76	....	162	104	36.3	95.5		35.3		37.0		

CALORIMETER LOG.

No. of run.	Duration of run.	Weight of exhaust steam.	Weight of water.	Radiation loss.	Quality per cent.*	Mean quality.
1	10 min.	1.40	.3	.07	86.5	88.6%.
2	10 "	1.62	.235	.07	90.1	
3	10 "	1.48	.287	.07	87.7	
4	10 "	1.80	.3	.07	89.1	

LOG OF TEST B. 27TH MARCH.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total mean H.P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	Mean H.P.	M. E. P.	Mean H.P.	
1	210	75	23.1	170	109	36.5	99	369 lbs.	35.8	4.32	37.1	4.36	8.52
2	218	80	23.1	170	109	36.5	99.5		35.9		37.4		
3	214	80	23.0	170	114	36.5	102.5		37.0		40.1		
4	218	..	22.0	170	114	36.5	104		36.0		37.8		
5	218	80	....	170	116.5	36.5	104		36.0		37.6		
6	210	75	....	170	120.5	36.5	106.5		35.0		38.1		
7	...	73	19.3	170	120.5	36.5	104		35.0		37.4		

CALORIMETER LOG.

No. of run.	Duration of run.	Weight of exhaust steam.	Weight of water.	Radiation loss.	Quality per cent.	Mean quality.
1	10 min.	1.60	.435	.07	82.1	82.1%.
2	10 "	1.40	.420	.07	80.2	
3	10 "	1.62	.395	.07	83.9	
4	10 "	1.55	.424	.07	82.1	

* The quality of steam is obtained from the formula

$$1 - X = (W - R) \div (W + \omega)$$

Where X = quality of the steam,

W = weight of water,

ω = weight of exhaust steam,

R = weight of steam condensed by radiation from calorimeter, and is found by experiment.

LOG OF TEST C. 28TH MARCH.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total mean H.P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	Mean H.P.	M. E. P.	Mean H.P.	
1	220	77	22.75	170	112	38	103.0	387 lbs.	35.4	4.43	35.9	4.25	8.68
2	222	77	21.3	170	121	38	107.5		35.2		37.0		
3	214	79	20.3	172	121	38	110.5		35.4		38.8		
4	220	80	....	172	117	38	110.0		35.0		36.8		
5	209	..	....	174	117	38	111.5		34.8		37.3		
6	216	78	21.3	174	122	38	113.0		24.8		35.5		
7	214	80	....	174	127	38	113.5		35.8		38.0		

Mean quality of steam, 77 per cent.

LOG OF TEST D. 28TH MARCH.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total mean H.P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	Mean H.P.	M. E. P.	Mean H.P.	
1	198	81	21	180	129	39	112	433.5 lbs.	33.2	4.22	35.6	4.13	8.35
2	210	81	21	176	120	39	108		33.2		35.6		
3	220	80	21.5	176	116.5	39.5	106.5		33.3		36.9		
4	214	80	21	176	112	40	103		36.6		36.8		
5	202	80	19	176	113.5	40	107.5		34.5		38.1		
6	210	82	19	180	112	40.5	108		38.2		38.0		
7	212	82	19	180	116	40.5	109.5		34.5		38.7		

CALORIMETER LOG.

No. of run.	Duration of run.	Weight of exhaust steam.	Weight of water.	Radiation loss.	Quality per cent.	Mean quality.
1	10 min.	1.50	.775	.07	69	68.6%.
2	10 "	1.75	.845	.07	70.1	
3	10 "	1.35	.760	.07	66.8	
4	10 "	1.50	.784	.07	68.6	

LOG OF TEST E. 29TH MARCH.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperatures.				Weight of condensed steam.	Head.		Crank.		Total mean H. P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	Mean H. P.	M. E. P.	Mean H. P.	
1	211	84	21.5	178	123.5	40.5	113.5	510 lbs.	36.8	4.30	39.2	4.25	8.55
2	192	....	20.5	178	123.0	40.5	113.0		37.0		39.8		
3	190	....	19.0	186	132.5	40.5	118.0		36.8		40.3		
4	196	86	19.0	186	133.5	39.5	119.0		37.2		40.6		
5	184	....	18.5	186	134.5	39.5	120.0		36.5		40.5		
6	204	....	20.0	186	128.0	39.5	115.5		37.2		40.2		
7	208	85	19.5	184	137.0	39.5	122.0		37.8		41.3		

CALORIMETER LOG.

Number of run.	Duration of run.	Weight of exhaust steam.	Weight of water.	Radiation loss.	Quality %.	Mean quality.
1	10 min.	1.45	1.187	.07	57.5	58.4%
2	10 "	1.55	1.182	.07	59.3	
3	10 "	1.37	1.125	.07	57.7	
4	10 "	1.47	1.130	.07	59.3	

LOG OF TEST F. 14TH APRIL.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total I. H. P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	I. H. P.	M. E. P.	I. H. P.	
1	193	83	20	202	140	49	98	306 lbs.	....	3.96	....	3.86	7.86
2	214	82	22	192	125	47	80		32		33.5		
3	217	62	24	160	104	47	80		31.8		37.1		
4	215	80	17	186	140	48	87		31.2		31.7		
5	218	64	22	168	132	48	67		31.9		30.9		
6	216	65	23	158	120	48	69		31.4		31.0		
7	218	60	24	156	105	48	69		30.4		32.3		
8	219	60	25	156	105	48	69		31.7		32.4		

Water per hour, 263 lbs.
Quality of steam, 99.15.
Dry steam per I. H. P. per hour, 33.1 lbs.

LOG OF TEST G. 18TH APRIL.

Number.	Revolutions per minute.	Steam-chest pressure, lbs.	Vacuum condenser, inches.	Temperature.				Weight of condensed steam.	Head.		Crank.		Total I. H. P.
				Exhaust steam.	Condensed steam.	Injection water.	Discharge water.		M. E. P.	I. H. P.	M. E. P.	I. H. P.	
1	216	60	25	150	100	48	93	356 lbs.	29	3.82	32.5	4.20	8.02
2	218	58	24.7	150	92	48	91		26		28.4		
3	220	70	24.5	154	96	48	94		28.6		32.5		
4	221	68	24.2	155	102	48	94		29.5		33.8		
5	222	68	24.5	157	104	48	90		31.2		34.8		
6	222	68	23.5	158	101	48	90		29.6		35.8		
7	220	68	24.5	157	101	48	90		29.7		33.5		
8	224	68	23.5	160	100	48	90		31.6		36 2		
9	224	65	24.5	157	101	49	90		30.9		35.2		

Water per hour, 267 lbs.
Quality of steam, 99.1.
Dry steam per I. H. P. per hour, 33.0.

The following table summarizes the results obtained from the log:

Test.	Quality of steam.	H. P. developed.	Weight of mixture of steam and water per hour.	Weight of dry steam per hour.	Dry steam per H. P. and per hour.
A	88.6	8.35	325	288	34.49
B	82.1	8.52	369	303	35.56
C	77.0	8.68	387	298	34.33
D	68.6	8.35	433.5	297.4	35.61
E	58.4	8.55	510	298	34.83
F	99.15	7.86	263	260	33.1
G	99.1	8.02	267	265	33.0

From this table a curve coördinating consumption of dry steam with quality of steam used has been plotted (Fig. 111) and shows that *throughout the range of qualities used the consumption of dry steam per indicated horse-power and per hour remains practically constant*, and would seem to indicate that, for the engine under consideration, the water was an inert quantity, doing neither good nor harm.

An attempt was made to apply Hirn's analysis to the experiments described. It was found, however, that if the usual assumption were made of dry steam filling the clearance space large errors would be produced in the result. These errors, due to unknown quantities of moisture in the clearance spaces, were sufficient to make the entire application untrustworthy and

of no value. The facts known regarding the exchanges of heat from steam to metal would also seem to make the results of this experiment appear reasonable, since it has been well established, that the water in steam rapidly increases during admission, that some evaporation occurs during expansion, but that at no time during the steam-stroke is the quality of steam in the cylinder equal to that in the main steam pipe; in other words, even when using dry steam, there is very much more condensation than evaporation. This being the case, it is difficult to see how the

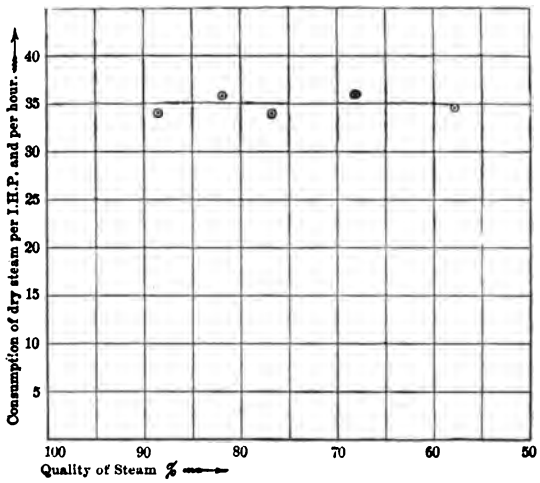


FIG. 111.

heat stored in any excessive amount of water can be utilized by conversion into work.

The following discussion of the results by Mr. L. S. Marks introduces several additional points of interest.

It is well known that with initially dry steam and unjacketed cylinders, water is always present in the cylinder except at the end of exhaust; so that extra water, if present, could not be evaporated till then, and hence its effect can only be to raise the back-pressure line.

The quantity of heat in the entrained water remains unchanged during admission; during expansion some of it is given up by conduction to the steam, and therefore raises the expansion line. The same, however, occurs during the exhaust, raising the back-pressure line.

The probable action of the heat in the entrained water is this: during admission it does no work; during expansion it increases the store of heat available for conduction to the steam and for evaporation of the water present. At the point of cut-off the heat available for these purposes is the sum of that stored in the cylinder walls and that in the water which is present, and an increase of this quantity will therefore result in raising the expansion line. This increase is maintained at the point of release, and results in an increased evaporation during exhaust, thus raising the back-pressure line also. For the purpose of corroborating this, the pressures at release and at beginning of compression and admission were measured on the indicator diagrams and are given below.

	Head.					Crank.				
	A	*B*	*C*	*D*	*E*	*A*	*B*	*C*	*D*	*E*
Release	23.5	26	26	25	29	28	31	32	32	33
Beginning of Compression	7	7.5	9	9	10	8	9.5	9	10	12
Beginning of Admission	40	45.5	49.5	46	50	41	46	48.5	48.5	49.5

As the admission pressure was nearly constant, these figures support the theory advanced above; they show that with decreasing quality of steam the pressures at the three points given increase. Hence, in explanation of the constancy of the consumption of dry steam, we are led to believe that the extra work done by the heat of the entrained water during expansion is sensibly equal to the extra negative work which it does during exhaust and compression.

In conclusion, the authors believe that this investigation shows that, for the class of engine under consideration, the heat carried in by the entrained water performs no useful function, and that a fair measure of the economy of an engine is the consumption of dry and saturated steam.

It is also thought that while the expression of economy in terms of B. T. U. per indicated horse-power will serve many useful purposes, the older standard of steam consumption should still be maintained.

DISCUSSION.

Mr. W. B. Levan.—I would like to ask the author, what is dry steam? I hear the words "dry steam" used.

Mr. Carpenter.—I do not know that I understand the inquiry. The term as used in the paper refers to *dry* and *saturated* steam, the properties of which agree with those in steam tables.

Mr. Frank H. Ball.—I think the conclusions of this paper are in the main reasonably correct, and I think they quite agree with the result of experiments conducted by Professor Denton, which, I believe, were reported to this Society; in which experiments he pumped into the steam pipe a known quantity of water, and found the same result which is reported in this paper. There is one point, however, which is not quite clear to me. I can very easily see how, after the admission valve closes, reëvaporation may change the position of the expansion curve; but it is not quite clear to me that, after the exhaust valve is opened to the atmosphere, any reëvaporation of water remaining in the cylinder could change the position of the lower line of the diagram. In other words, it does not seem possible to me that the capacity of the exhaust pipe could be overtaxed by the passage through it of the steam which may be evaporated from water remaining in the cylinder. I would like to have Professor Carpenter explain that point.

Prof. D. S. Jacobus.—Mr. Ball is correct. Professor Denton's experiments, which were presented in 1888 to the American Association for the Advancement of Science, are to be found in the report of their transactions for that year. The tests were conducted by pumping a known weight of water into the steam pipe. It was found that, when the cooling effect of the water was allowed for, the dry steam per hour per horse-power remained unchanged.

Professor Carpenter said something about the peculiar action of superheated steam. We have made experiments by admitting hot water into the steam pipe of a compound engine where the steam was superheated from 5° to 20° Fahr. These experiments were made on the Pawtucket Pumping Engine, and may be found in the discussion of Mr. Kent's paper in Vol. XIII. of the transactions. We found that with 17° of superheating the weight of water per hour per horse-power was about 1½% less

than with steam containing a small amount of moisture, so that there was no remarkable action due to the superheating.

Mr. David L. Barnes.—I have read the experiment described in Mr. Carpenter and Mr. Marks' paper, on the effect of water in steam on the economy of the steam engine, with much interest, and I think the conclusions reached are correct for the apparatus used. It should be remembered, however, that this engine was a very small one, and that while in the most extreme case there was about 42% of water in the steam entering the cylinders, there was a further amount of water added before cut-off by condensation, of nearly 50% of the steam which entered. This I have learned by an examination of indicator cards (Fig. 110). The water per horse-power per hour measured from the indicator cards is not far from 18, while the actual water rate was 35 lbs. This experiment, therefore, proves mainly that, under the conditions of a small engine with great initial cylinder condensation, a large amount of water entering with the steam produces but little, if any, effect on the economy of the engine. This might be expected from the conditions, which are quite unusual. If the same experiment was performed with a larger engine, or with a compound engine, the resultant line shown on diagram Fig. 111 would be materially different. There is, in the engine tested, but little chance for reëvaporation, and little useful effect is produced by the heat in the water which passed into the cylinders with the steam. With greater expansion the heat in the water would be utilized to a greater extent, and without charging up the entering water to the engine, as is the case with diagram Fig. 111, the resultant line would incline upward to the right, and the consumption of dry steam per indicated horse-power per hour would decrease as the percentage of water in the entering steam is increased. It is interesting to note that the indicator cards in Fig. 110 have decidedly different characteristics. The one from the head end shows condensation after cut-off, and little if any reëvaporation except at release, while the one from the crank end shows considerable reëvaporation during expansion. The compression line on the card from the head end shows a considerable increase in steam weight during compression, while on the card from the crank end, the compression line approximates closely to the rectangular hyperbola. On both cards there is a very unusual change in the compression curve near the point of ad-

mission. It is a shape of diagram which I have never seen before and do not understand. It may be that the indicator cord was too long or there was some defect in the mechanism, but I do not suppose this was the case. I am inclined to think it is due to some unusual condition arising from the small size of the engine. The conclusions of this paper should not alter the opinion, which I think many have held, that the consumption of *dry steam*, entrained water not included, per indicated horse-power per hour will be decreased when considerable water passes into the cylinders with the steam; but of course, the efficiency of boilers and steam engines together is less somewhat in proportion to the per cent. of water entering with the steam.

Mr. Frederick A. Scheffler.—I would like to ask Professor Carpenter if this engine had the valve especially made with the eccentric accompanying it to give it the motion, as I see that it was a plain slide valve engine with a throttling governor. I have never seen an engine which had a throttling governor and a plain slide valve on it show a card with such a short cut-off.

R. H. Thurston.—The facts which have been so well exhibited by the investigation which Professor Carpenter describes are, in a general way, well known, and were familiar to engineers many years ago. Combes, who as long ago as 1843 proved formally what Carnot had indicated to be the probable fact, the thermo-dynamic liquefaction of steam in adiabatic expansion, also showed this now rediscovered fact to be true. As I have been in the habit of stating in my lectures for years past:

"Where the ratio of expansion is not great, the presence of the entrained water in the steam may not produce any ill effect. For the ideal case of the non-conducting cylinder, this was long ago shown to be true by Combes.* This deduction was also found to be correct in practice by Hirn. With large ratios of expansion, and especially with jacketed engines, the opposite is probably always true." †

It is easy to see that, in the pure thermo-dynamic case, the action of water in the mass of working steam could have little effect upon the economy of the operation of the engine. Its effect would be to merely supply a small amount of additional heat for partial utilization by conversion into work. The real

* *Théorie Mécanique de la Chaleur*, chap. xxxv., p. 157.

† *Manual of the Steam-Engine*, Vol. I., chap. v., p. 524.

question which interests us, from the point of view of the practitioner, is: What variation will the presence of water in varying amounts produce in the magnitudes of the internal wastes? This can only be settled by such experiments as these. Where the steam enters moist or wet, if the expansion is considerable, I can imagine its having a very observable effect, and especially with jacketed engines, where the work of the jacket in drying the steam, an essential to its real purpose of preventing initial condensation, may thus be sensibly, perhaps sometimes greatly, increased, and an otherwise efficient jacket thus rendered ineffective. Every pound of water entering the cylinder and evaporated by the jacket during expansion, and particularly during exhaust, is a tax upon the economical value of the machine. It is extremely desirable that this investigation be continued, and with jacketed and high-expansion engines.

A serious difficulty which must, perhaps, always be met, and which unquestionably gives uncertainty to all experimental work thus far reported, rendering their conclusions entirely unsatisfactory on that score, is the apparent impracticability of insuring the mingling of the water present with the steam in such manner as to produce a homogeneous mass; such as probably comes over from the boiler when its steam is wet within the usual limits of regular working, and until intermittent "priming" begins. A stream of water introduced, as in some experiments, into the steam-pipe, may pass as a solid stream through the engine, and have little or no sensible effect in exaggerating either thermo-dynamic or thermal wastes. Just what difference, if any, this fact would make, no one can say without investigation.

*Prof. R. C. Carpenter.**—In answer to several of the inquiries, I would say, first, in regard to the engine, that it is one built specially for experimental purposes. We have six sets of valves, and we can vary the cut-off as desired by changing the valves. This accounts for the early cut-off shown in the diagrams. In every other respect it is a commercial engine. In regard to the inquiries by Mr. Ball, I do not know that I am able to answer them fully and satisfactorily. I can state the facts of the case; the reasons will no doubt appear later. Two reasons, both suggested in the paper, would account for a portion, if not all, of

* Author's Closure, under the Rules.

the increase in back pressure. These are reëvaporation and increased work on the back stroke; the latter of which I consider responsible for most of this increase. The water in the steam, when in such large quantities, has to be lifted out of the cylinder by the piston. This work is no doubt in all cases an appreciable quantity, and I can readily see how it might equal or exceed any gain due to evaporation of water contained in the steam. In the engine in question the valve is on the side of the cylinder, the port openings are large, and the work done in removing the water cannot be unusual. These are the only reasons that I can give or offer for the increase in the back pressure; and whether these are sufficient for a complete explanation, I am not certain. All I can say is that the results given are simply actual measurements of the diagrams taken. In regard to Mr. Barnes's observations, what he says regarding application of these results to engines of a different class is no doubt perfectly true. He says that the case would be very different with a compound engine. Now, I cannot see any reason for that assertion, nor does he give any reason for his belief. I have made a great many experiments where we have measured the heat losses in compound engines from the beginning through, and, as a rule—I may say a universal rule for unjacketed engines, because I have never known of any case to the contrary—the quality of steam in the low-pressure cylinder has been less than in the high-pressure cylinder. If we were to start with dry steam in the high-pressure cylinder and work that steam to a lower pressure through two cylinders, our resultant quality at the end of expansion in the low-pressure cylinder would not be higher than if we had worked it through the same temperature ranges only in one cylinder. I am quite certain from a study of available analyses of tests which have been made where the heat has been measured in compound engines, that such a statement is true. I do not see how it is possible for this reëvaporation of steam to do much useful work, when the resultant loss due to lifting out the water is considered. It may be that under some circumstances the extra heat in the entrained water does do useful work; but I cannot readily see how it is possible, especially if it be greatly in excess. I may also say that I was pleased to learn that the experiments made by Professor Denton showed the same general results as stated here. I believe they were made on the same general class of engine, and experimentally at least

the question is still open as to the effect of water on the economy of compound or triple expansion engines.

The opinion expressed by Mr. Barnes, "that the heat taken into the engine by entrained water might add to the store of useful work in a compound engine," seems to be quite contrary to that usually held by engineers respecting the effect of water in steam. While it is not, perhaps, worth while to spend much time in obtaining what is merely the expression of prevalent opinion on this subject, yet no doubt it may be a matter of interest, since on such a basis rests the principal part of the discussion regarding the matter.

I will merely refer to one or two cases: *Transactions A. S. M. E.*, Vol. XII., page 655, report on test Triple Expansion Engine, by E. D. Leavitt, Jr., and J. T. Henthorne; also see more complete report published by Edw. P. Allis & Co. After a statement that the water had been deducted, the following remark was made:

"It is my opinion that 7.39% of water in the steam with which it is mixed would make a difference of at least 10% less duty than would be performed with dry steam, or not less than 2½% after subtracting the weight of water itself."

Mr. W. C. Kerr informed the writer that he had made an extensive canvass of noted engineers regarding opinions held on the subject, and in every case the expression was, that the entrained water had a detrimental effect on economy, in excess of the correction for the weight. (See also his lecture before Franklin Institute on "The Steam Loop.") In a recent editorial in the *Engineering Record*, the same expression is forcibly stated.

It is quite true that the experiments in this line cover a limited field in steam-engine practice, yet it is interesting to note that the results of the experiments indicate that the effect of water in the steam on the economy of the engine is a mean between opposing views said to be entertained respecting the matter.

The following table gives some information regarding the interchanges of heat, which it is believed will be of interest as bearing on some of the points raised during the discussion. The results were obtained by computations from measurements taken during the test by Mr. L. S. Marks; the computations serve merely to show that the heat taken into the engine by the entrained water did not increase the useful work.

Name of Test.	*A.*	*B.*	*C.*	*D.*	*E.*
Quality of steam entering.....Per cent.	88.6	82.1	77.0	68.6	58.4
Quality of steam at cut-off.... ..... "	52.2	46.5	47.8	41.7	34.8
Quality of steam at release ..... "	62.6	58.2	57.7	49.4	43.7
Entrained water entering..... "	11.4	17.9	23.0	31.4	41.6
Entrained water at cut-off ..... "	47.8	53.5	52.2	58.3	65.2
Entrained water at release..... "	37.4	41.8	42.3	52.6	56.3
Condensed during admission ..... "	36.4	35.6	29.2	26.9	23.6
Reëvaporated during expansion .... "	10.4	11.7	9.9	5.7	8.9
Steam condensed per revolution *admission*, Lbs.	0.00905	0.01018	0.0085	0.0092	0.01005
Steam reëvaporated per revolution *expansion*, Lbs.	0.00258	0.00335	0.00295	0.00196	0.00382
Heat in one pound wet steam.....B. T. U.	1,078	1,022	976	903	815
Heat required per I. H. P. (from 152°) per minute .....B. T. U.	615	650	637	678	692
Ratio heat required per I. H. P.....	1.00	1.056	1.037	1.10	1.122
Heat provided by entrained water...Per cent.	3.03	5.14	6.91	10.2	14.
Heat per I. H. P. per minute in entrained water.....B. T. U.	18.5	33.5	44.2	69.0	101.0
Heat per I. H. P. per minute less that in entrained water at admission.....B. T. U.	597	617	593	609	591

DLXXIV.*

CONSTANTS FOR CORRECTING INDICATOR SPRINGS THAT HAVE BEEN CALIBRATED COLD.

BY R. C. CARPENTER,† ITHACA, N. Y.,
(Member of the Society),
LIONEL S. MARKS, BIRMINGHAM, ENGLAND,
AND BY S. H. BARRACLOUGH, SYDNEY, AUSTRALIA.

OBJECTS OF TEST.

THE object of the following investigation was to determine the average percentage difference in the readings of indicators when tested hot and cold.

Incidentally, also, the method used gave the calibration of the springs tested, and indicated whether they were originally calibrated hot or cold. It also gave the magnitude of the friction of the moving parts of the indicator.

The investigation does not, however, make any comparison of one indicator with another, nor do any of the data taken permit us to draw any conclusions as to the relative accuracy of the various styles of indicators. The errors which were found were more often individual than characteristic of a series of similar indicators, and, consequently, cannot be used as a basis of judgment on which to form an opinion of relative merits.

It may be stated that these errors are of sensible magnitude in nearly every case, and the investigation would seem to indicate that it is practically impossible to construct a spring and accessory moving parts of an indicator so as to give a diagram perfectly accurate for any prescribed scale.

It was, in general, noticed that new indicators gave diagrams agreeing more closely with the scale than the old ones. Whether

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* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† The investigations described in the following paper were conducted at my request by Mr. Marks and Mr. Barraclough, and the following report was largely written by them.

R. C. C.

the reason for this is to be found in improved processes of manufacture, or in the fact that the diagram ordinates are affected by wear of parts, or by some change in the temper of the spring, was one that could not be determined by the authors.

As is doubtless well known, two methods of calibration are in common use, in one of which the scale of the diagram is compared with the reading of a weighing-scale subjected to the same stress, or with a definite amount of standard weights, the temperature of the spring being that of the surrounding room. This we term the cold calibration. The other requires the use of an accurate manometer, which receives the same steam pressure as the indicator piston; the diagram is compared with the readings of the manometer, and in that manner the accuracy of the scale is determined.

The present practice among indicator manufacturers seems to be varied as to the method used. At least it is quite certain that of the indicators which have been sold the last few years some are more nearly true to scale when calibrated cold, while others are more nearly correct when calibrated with steam.

METHOD OF TESTING.

The method of testing was to compare the readings of a very accurate manometer with the ordinates of the indicator diagram, when subjected to steam pressure for the hot test, or air pressure for the cold test.

More or less experimental work relating to the methods of calibrating indicator springs has been in progress the past three years in the laboratories of Sibley College. Both cold and hot tests have been thoroughly tried. The first methods attempted of calibrating the springs with steam pressure utilized a large vessel, in which the steam pressure could be regulated by throttling either admission or discharge valve. This vessel was put in direct connection with the indicator piston and with an open mercury column. The column has a glass tube about 25 feet in length and $\frac{1}{2}$ inch in diameter, with an attached scale on wood. This method, although used for some years, was found not to be very delicate, and it was almost impossible to secure the same results in the calibration of the same spring at different times. As this difficulty was found not to pertain to the method of regulating steam pressure, it was at first believed to be due to changes in the scale of the indicator springs, due to use.

Subsequent investigations showed pulsations of a wave-like character in the mercury, which continued for a long time.

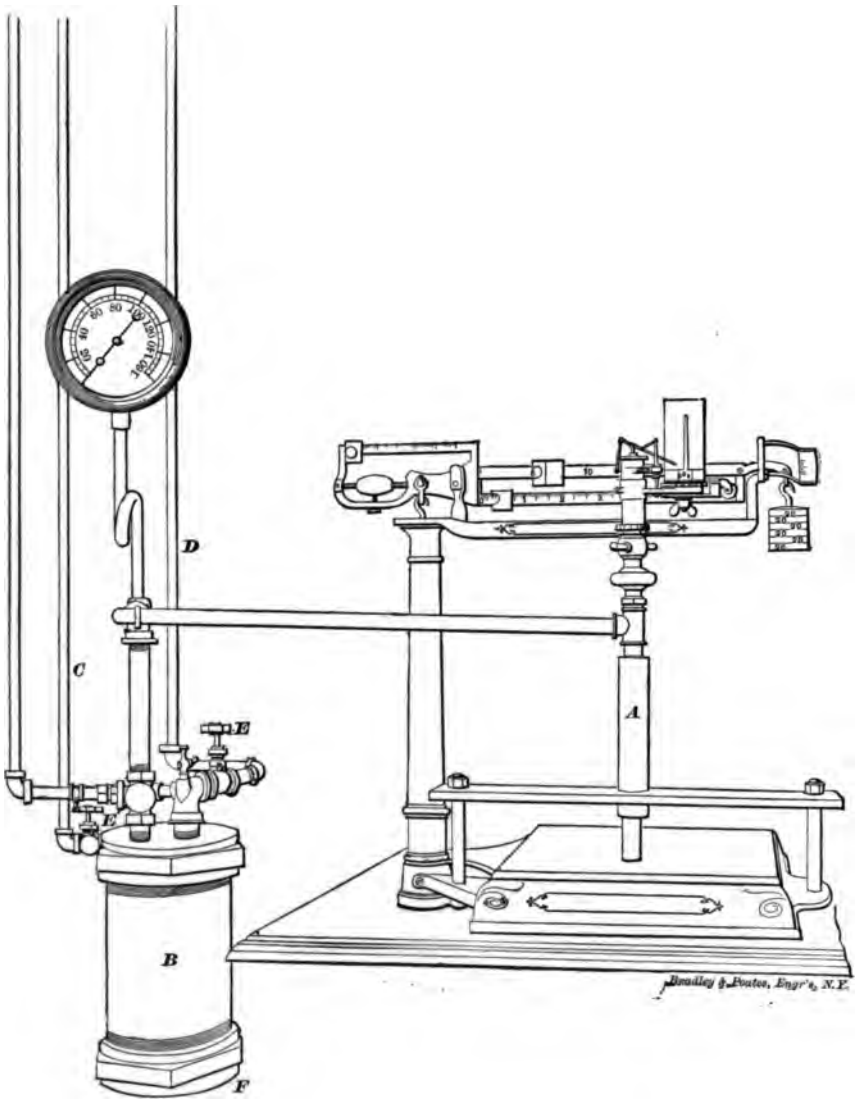


FIG. 112.

While the amplitude of these pulsations was not great, it was sufficient to obscure delicate reading, and tended to produce those

variations in the results which had been formerly ascribed to irregularities in the indicator spring.

Prof. M. E. Cooley's method of testing indicator springs at ordinary temperature was then instituted. This method briefly was to compress the indicator spring by pressure on a rod, the reactionary force being measured by delicate weighing scales. This, so far as delicacy and uniformity of results was concerned, proved all that could be desired; but doubt as to the effects of the temperature led to the substitution of the apparatus shown in Fig. 111 and Fig. 112.

In this case a piston arranged so as to be readily rotated and reacting on a weighing scale was used as a manometer. With this apparatus delicate and uniform results were obtained.

DESCRIPTION OF APPARATUS.

The apparatus first used is shown in the accompanying print (Fig. 112). A cylinder (*A*) is screwed on to a cross-bar firmly supported above a Buffalo scale which reads accurately to $\frac{1}{50}$ of a pound. The length of the cylinder is 8 inches, its diameter 1.125 inches; a piston 1.123 inches in diameter, or with area equal to 0.993 square inches, working freely in the cylinder, rests upon the scale. The indicator to be tested is screwed into the top of the cylinder.

B is a small reservoir communicating, by pipe *C*, with the boilers, which are distant about 200 feet, or, by pipe *D*, with a reservoir of compressed air. The pressure in the air reservoir was maintained automatically by a Westinghouse air-pump. A pressure gauge was inserted in the reservoir, and also two pipes, one leading to the atmosphere, the other to the cylinder *A*, and the indicator.

A form of the apparatus, more compact and convenient for use, which was used in the later experiments, is shown in Fig. 113, as manufactured by Schaffer & Budenberg, N. Y.

In this case the manometer piston, *K*, rests directly on the yoke, *N*. The pressure is transmitted and balanced by a scale beam sensitive to $\frac{1}{50}$ of a pound.

During the operation of weighing the steam pressure the piston is kept in partial rotation by moving a rod attached to pin, *P*.

Steam is admitted at *H*, discharged at *I*.

METHOD OF USING APPARATUS.

The springs were tested hot with steam, and cold with compressed air. This latter the authors believe to be an innovation of some importance, since the conditions of the two tests are by

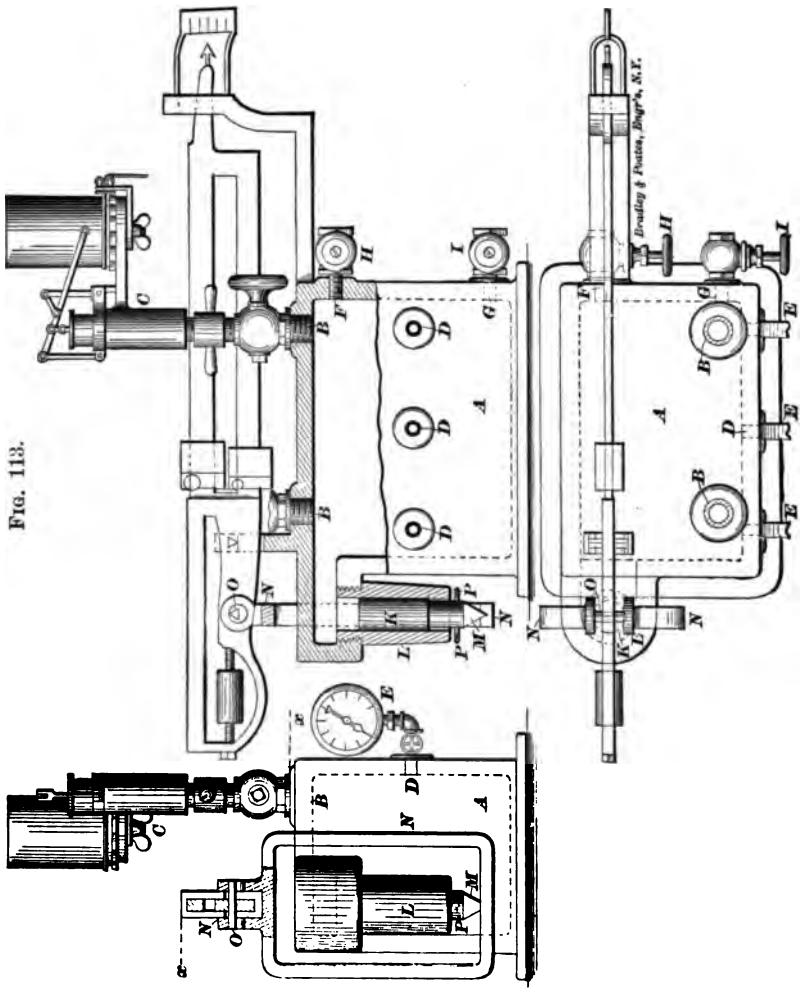


Fig. 113.

this means made precisely similar, with the exception of temperatures. Both pistons were oiled before starting, with white spindle oil for the cold tests, and with cylinder oil for the hot. In the latter the indicator was warmed by blowing steam

through before starting. The scale was adjusted to zero reading, with the piston resting upon the platform, the piston being lightly tapped or rotated to prevent any tendency to stick, the same precaution being taken throughout the whole of the investigation.

The method of making a test was as follows:

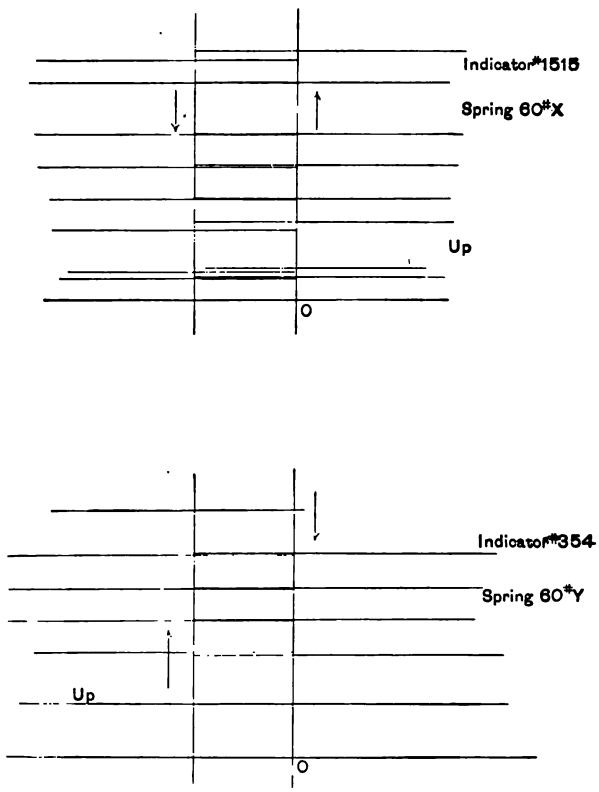


FIG. 114.

The poise was set at a given load, and the pressure of steam or air was allowed slowly to rise, by adjusting the inlet and waste cocks. At the instant when the pressure had risen so as to make the beam float in its mid position, a line was drawn across the drum. The poise was thus set at successive increasing loads till the desired maximum was reached; then, after increasing the pressure beyond this maximum, the same loads

were traversed in the opposite direction and corresponding lines drawn.

The cards were similar to those shown in Fig. 115. All the lines were drawn between the two vertical lines, those taken with rising pressures being extended to the left, those with falling pressures to the right.

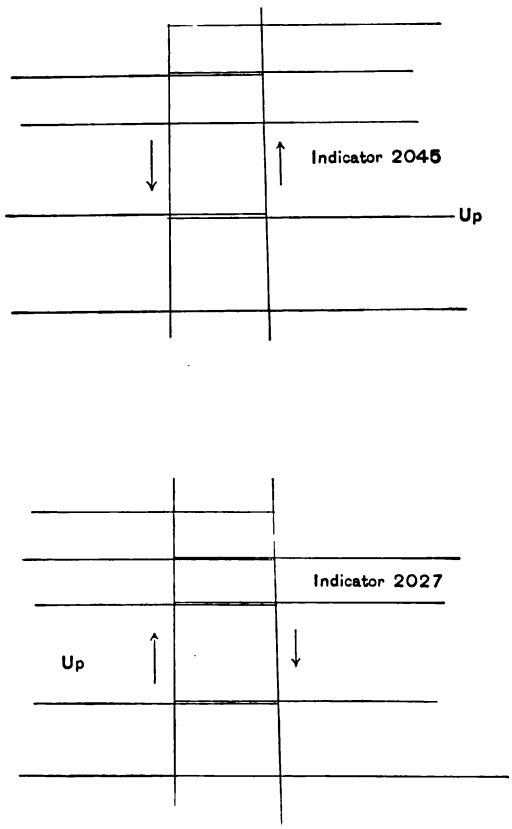


FIG. 115.

The diagrams actually taken during the calibration of the indicators used on the test of the North Point Station at Milwaukee, Wis., are shown in Fig. 115, the calibration curves for each case being given below.

The relative errors are then drawn on a large scale by laying off errors as ordinates and total pressure as abscissa, as shown in Fig. 116.

GRAPHICAL RESULTS OF TEST.

The results of these different tests are shown graphically in the accompanying diagrams, Figs. 118 to 126.

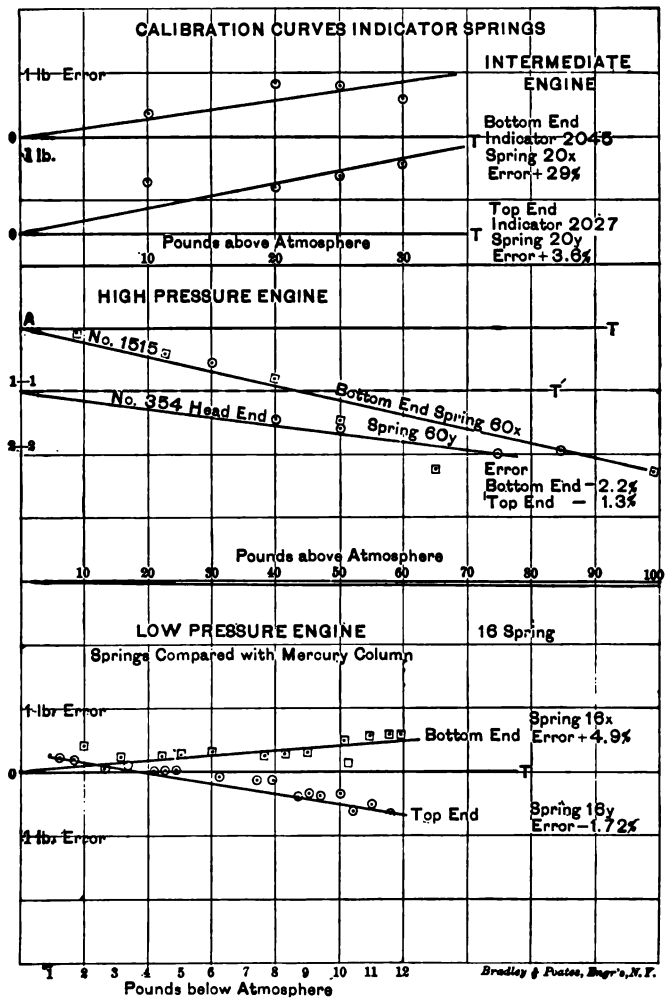


FIG. 116.

The errors of mean reading for each spring are co-ordinated with the pressures per square inch, corresponding to those errors, for both hot and cold tests.

The calibration curve of the hot test is shown by a full line, that of the cold test by a broken line. The line of reference in

each case is marked 0-0, and the tangent of inclination of the curves to the reference line is a measure of per cent. of error.

The relative scale of error is very much magnified, but it will be noted that the variation of the observations from a right line

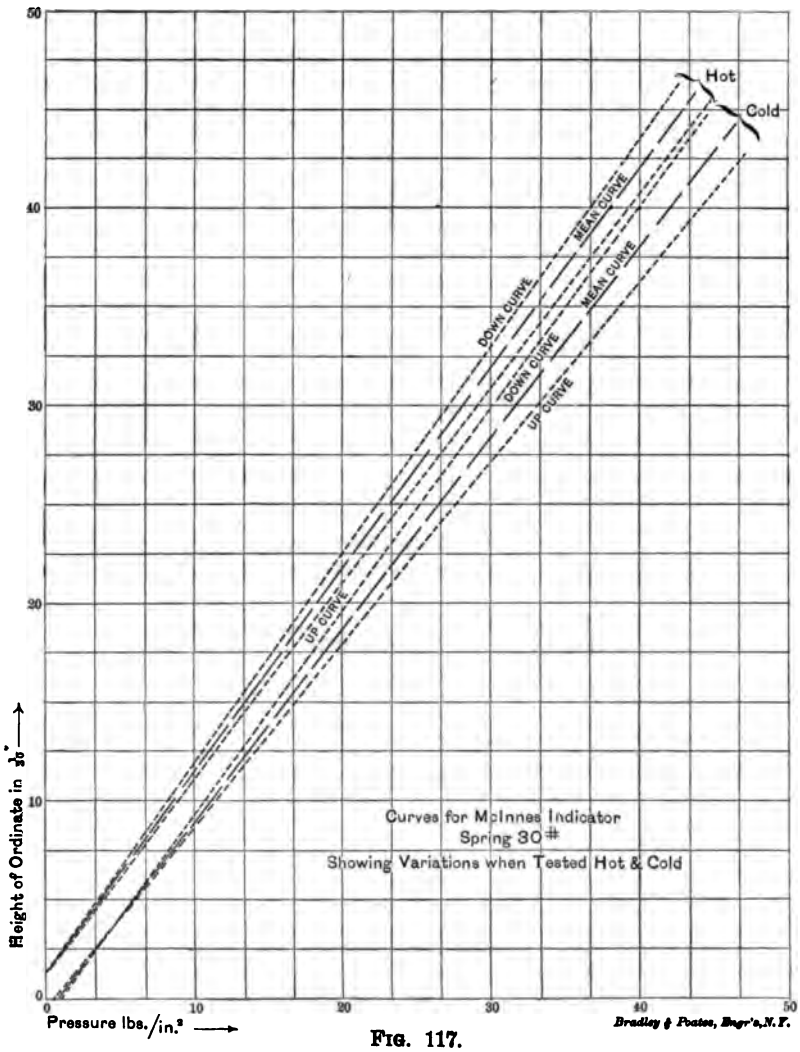


FIG. 117.

is very small, and generally less than errors in measuring the ordinates of the diagram.

If the calibration lines were horizontal, the diagrams would show that the error of the spring was zero for all parts of the

scale; or, at least, that the error would not affect the length of ordinate of the diagram; if the calibration lines intersect the 0-0 line at the origin, the diagram is correctly located with reference to the line of zero pressure or the atmospheric line.

In general, failure of these lines to intersect at the origin can be explained as due to lost motion and friction in some of the moving parts of the indicator.

This variation is actually small, in every case, although apparently large in some cases, due to the magnified scale of error used in constructing the diagrams. The instruments were in every case in good order, and the lost motion not apparent on ordinary investigation.

RESULTS OF THE TEST.

The indicators chosen for investigation were:

The Thompson, Crosby, McInnes, Elliott's "Darke," Buffalo, Tabor, Bachelder, and Calkins.

In all, thirty-two springs were tested. At first two cards were taken for each set of conditions, but finding that the results were uniform it was decided that one card was sufficient for each case.

The observations and results of the tests are given in the accompanying log.

In the log the third column, headed "Scale of Spring," gives the maker's calibration of the particular spring.

The next column gives the pressure on the scale, and since the area of the piston is .993 square inches, this is equivalent to the pressure per square inch given in the following column.

Columns 6, 7, and 8 give the heights of the ordinates of the lines drawn on the up and down grades, and the mean of these respectively for the cold test. The error in height of ordinate for any pressure is due to the friction of the indicator. This friction opposes the motion of the piston so as to make it less on the up grade and greater by an equal amount on the down grade. Hence the difference of the heights of the ordinates for the same pressure on the up and down grades is a measure of the friction of the indicator, being equal to twice that quantity. The friction thus obtained is the sum of that between piston and cylinder, and that of the pencil motion, but does not include the friction between pencil and paper, as the pencil was always held away from the drum except when drawing an isopiestic line. In

column 9, the average friction is obtained by summing the differences of the heights of the down and up ordinates for each pressure, and taking one-half of the mean of these quantities.

The friction can be obtained also from a diagram co-ordinating pressures and pencil movements, as shown for the McInnes indicator in Fig. 117. If this friction is constant these straight-line curves will be parallel to one another for up and down grades. This was not usually the case, the lines often being inclined to one another at a small angle. It was found that the mean friction obtained from such a diagram does not differ appreciably from that obtained from the data as above.

The column headed "Error of Mean Reading" is obtained by subtracting the actual pressure per square inch, as given in column 10, from the height of mean ordinate given in column 8. Columns 11–15 have same significations as columns 6–10, but are for hot tests.

The last column gives average percentage difference of readings for the same pressures when hot and cold, obtained from the graphical representation of the results of the test as explained below.

In conclusion, the results of this investigation demonstrate that the practice of calibrating indicator springs cold leads to an error on the average of about 3.6%. The proper method to be pursued in any case before using an indicator is, of course, to calibrate it with steam; but if it is only practicable to calibrate it cold, or if the maker's cold calibration has to be used, a reduction of the M. E. P. obtained by 3.6% will generally give a value correct within 1%.

It will be noted that this investigation only considers the condition for constant steam pressure; while in actual use in the engine the pressure varies rapidly.

Since indicator springs are not compressed beyond the elastic limit in good practice, there is no apparent reason why Hookes' law for an elastic body should not apply, in which case the percentage error which was found true for a static calibration should hold correct for the case of varying steam pressures, such as occur in the actual working of the engine. To find out whether the static calibration did apply, the following experiment was performed: two indicators were selected, and the springs were calibrated with steam pressure on the instrument shown in Fig. 113, and in the manner already described. The result of this calibration is shown in the accompanying diagram, and, when measured,

was as follows: Indicator K scale of ordinates of diagram 0.13% too great; indicator T scale of ordinates 2.26% too great; difference by static calibration, 2.13%. The two indicators were then connected by means of a Tee fitting, to the same end of a Corliss engine cylinder. Steam connection was made, and a simultaneous diagram taken from each indicator.

A careful integration of each diagram, supposing the scale in each case to be correct and equal to 40, shows a mean effective pressure of 27.04 for the diagram taken with indicator K, and of 26.50 for the other. Increasing these results by the amount of correction that the static calibration indicated as required, we have the following results:

Corrected M. E. P. indicator K', $27.04 \times 1.0013 = 27.04 + 04 = 27.08$.
Corrected M. E. P. indicator T', $26.50 \times 1.0226 = 26.50 + .59 = 27.09$.

While these results are not identical, they are sufficiently close to show that the differences found by static pressures are substantially the same as those obtained under the actual conditions of operation, and furnish a strong argument for establishing the reliability of the method described of correcting the mean effective pressure of the indicator diagram, by the factor shown to be necessary by the static calibration under steam pressure.

DISCUSSION OF RESULTS.

A study of the results of the test indicates that errors of indicator springs are of such magnitude that they cannot in general be neglected.

Secondly, these errors are in proportion to the pressure, and consequently the correction for the same is a percentage to be added to or subtracted from the mean ordinate. There are probably some exceptions to this general law, and no doubt individual springs may show an irregular error, due to change in friction as the piston changes its position, as well as in variation in the elasticity of the spring itself. This point is discussed later.

The springs tested were, however, of a great variety of make, and the calibration curves are practically straight lines in every case.

The loads applied to the springs were well within the elastic limit; had that been exceeded, the error would probably rapidly increase.

From the calculation given later, it would seem probable that

the temperature of the indicator spring does not vary a great deal, due to change of pressure of the steam; it is certain that the steam which escapes past the spring is not sensibly higher than that due to atmospheric pressure.

The two calibration curves for each spring would both pass through the origin of co-ordinates if friction and lost motion did not interfere with the position of the no-pressure line, for in that case the error of mean reading would be zero. The effect of friction, however, is to shift the curves parallel to themselves, so that they frequently do not pass through the origin. The difference of the tangents of the inclinations to the horizontal axis of the hot and cold curves, for any spring, multiplied by the ratio of the scale of ordinates to the scale of abscissæ, gives the average percentage difference of readings for that spring when hot and cold.

Collecting the results for the different indicators it is seen that the average percentage difference of readings, hot and cold, varies from 2.5 to 6.6, and that the mean is 4.1% As, however, the Elliott indicator, owing to a tightly fitting piston, gives a very high average percentage difference, and the Bachelder differs in type from the rest, it appears to the authors to be wiser to neglect the results obtained from these two indicators, so that the variation is from 2.5 to 4.4, and the final mean average percentage difference between the cold and hot test becomes 3.6%.

DISCUSSION OF THEORY OF TEMPERATURE EFFECTS.

The chief causes operating to produce variation in deflection of an indicator spring with temperature are changes of:

(*a*) Area of piston.
(*b*) Elasticity of spring.
(*c*) Length of spring.

(*a*) All the indicators tested (with one exception) had brass pistons, and these, being in contact with the steam, would approximate its temperature. The linear expansion of brass is given by the formula:

$$L' = L\,(1 + .0000104\,t),$$

where L and L' are lengths at 0° and t° Fahr. respectively.

The experiments were carried on with steam pressures varying from 0 to 80 lbs. above atmospheric pressure. These would give ranges of temperature of the piston from 60° Fahr., when

tested cold, to from 212° to 320° Fahr. when hot, *i. e.*, of from 152° to 260° Fahr. The corresponding changes in diameter of piston are given by

$$\frac{D'}{D} = 1 + .0000104 \times 152$$
$$= 1.001581$$

and
$$\frac{D''}{D} = 1 + .0000104 \times 260$$
$$= 1.002704$$

or of areas
$$\frac{A'}{A} = 1.0032$$

$$\frac{A''}{A} = 1.0054.$$

Hence the total pressure on the spring is increased in ratio of from 1.0032 to 1.0054, with increase of temperature of piston from 212° to 320° Fahr.

(*b*) The effect of increase of temperature on the elasticity of a steel spring has previously been investigated by Mr. L. S. Marks, at Mason College, England, who found it to be represented by the formula—

$$E' = E(1 - .00019\,t),$$

where E and E' are moduli of elasticity of the coiled spring at 0° and t° Fahr., respectively.

It is probable that in actual practice the temperature of an indicator spring never rises more than a few degrees above 212° Fahr. If, within the range of high pressures commonly in use, the quality of steam is above .95, the steam escaping past the indicator piston under atmospheric pressure would be superheated; but it is fairly certain that in the great majority of engines the steam does not attain this quality except for a short period of the stroke, near the end of exhaust. As the steam is then at a low pressure its quality can rise considerably above .95 without superheating taking place, and under any circumstances the superheating is small and is only effective for a short period of the stroke, so that the result will be to raise the tem-

perature of the escaping steam only a few degrees above 212° Fahr.

The effect of conduction of heat from piston to spring tends to raise the temperature of the latter to a considerable degree, unless the escaping steam is wet, in which case a film of water will be deposited round the spring, and heat conducted very rapidly from it.

Radiation from the cylinder walls will also be effective to a slight degree in raising the temperature of the spring.

If we assume 220° Fahr. to be the mean temperature of the spring in common practice, we find that its elasticity is diminished to

$$E' = E\left\{1-.00019\,(220-60)\right\}$$
$$= E \times .9696,$$

and since the deflection of a spring is inversely as its elasticity, the deflection for the same pressure is increased in ratio $\frac{1}{.9696}$, or 3.13%.

(*c*) A spiral spring, on heating, uncoils and elongates in the direction of its axis. If it maintained, when hot, the elasticity which it had when cold, it would under the same pressure compress the same fraction of its length, and as this length is increased, the deflection is increased in same proportion. This effect, however, is of the second order of magnitude, and so can be neglected.

The combined effect of the causes (*a*) and (*b*) is to increase the deflection of the spring in the ratio of the product of their separate effects. This gives a percentage increase of deflection varying in ordinary practice from 3.45% to 3.66%, with increase of steam pressure from 0 to 80 lbs. per inch above atmospheric pressure.

The close agreement of this, calculated with the experimental results, tends to prove that the constants assumed agree substantially with those occurring in practice, and would seem to indicate that they might be used with confidence.

INDICATOR H.

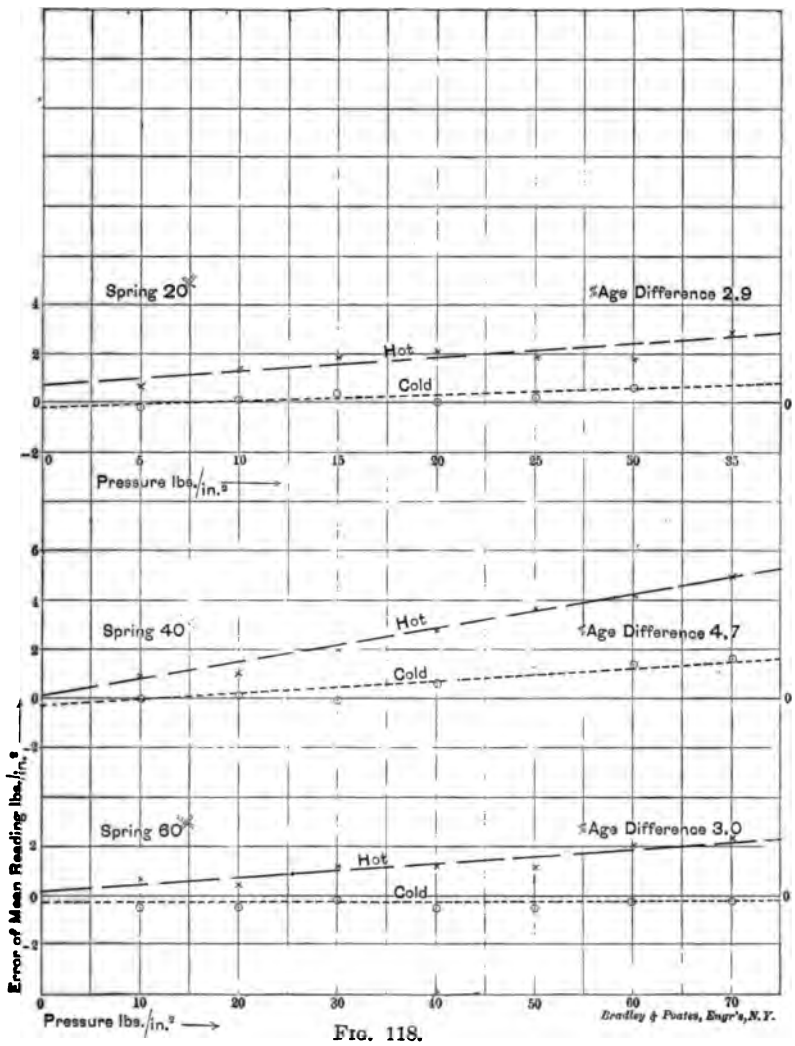


FIG. 118.

INDICATOR F.

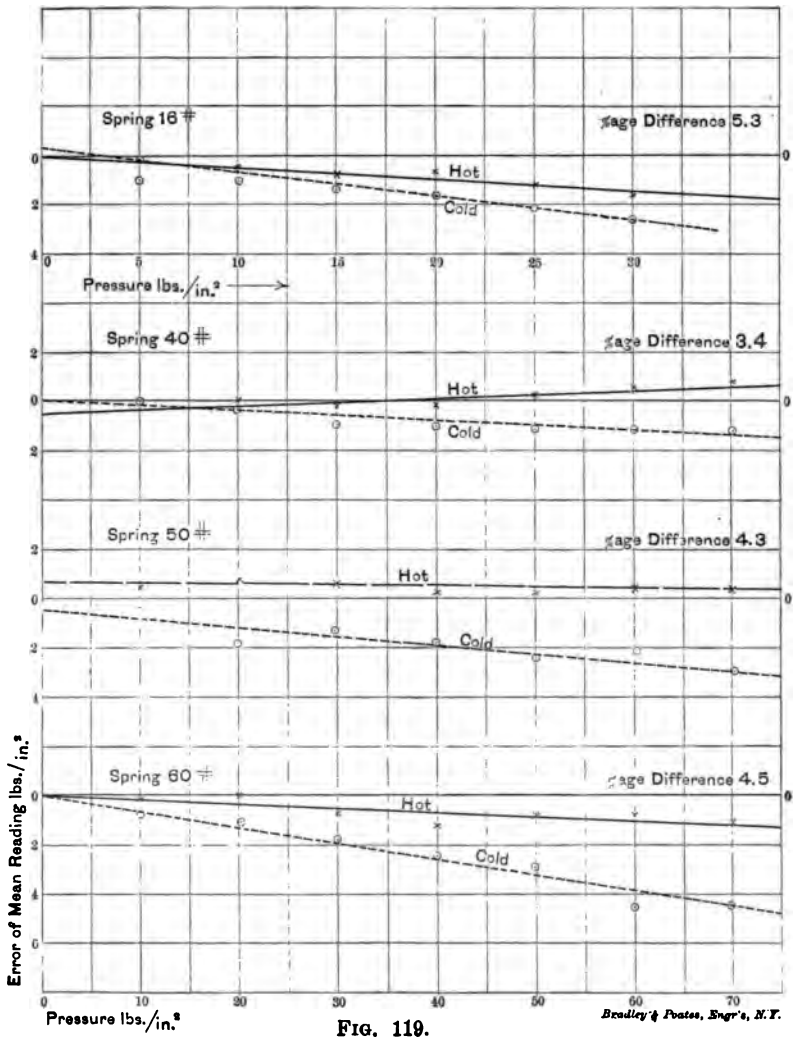


FIG. 119.

INDICATOR D.

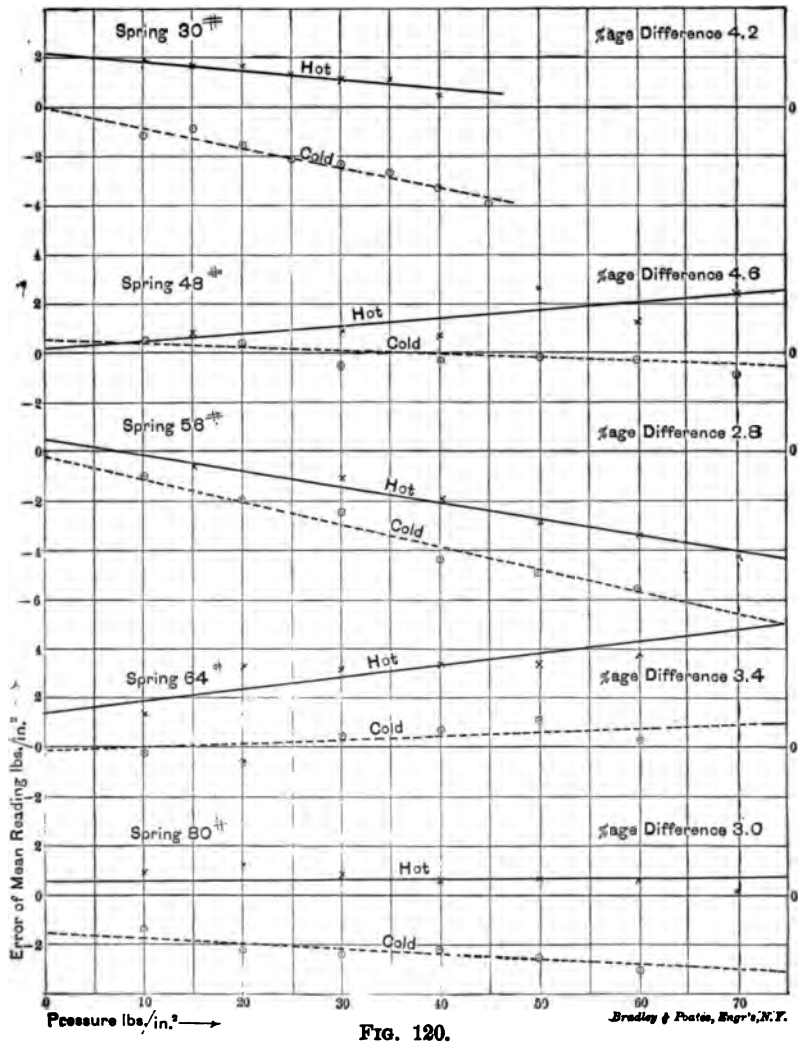


FIG. 120.

INDICATOR B.

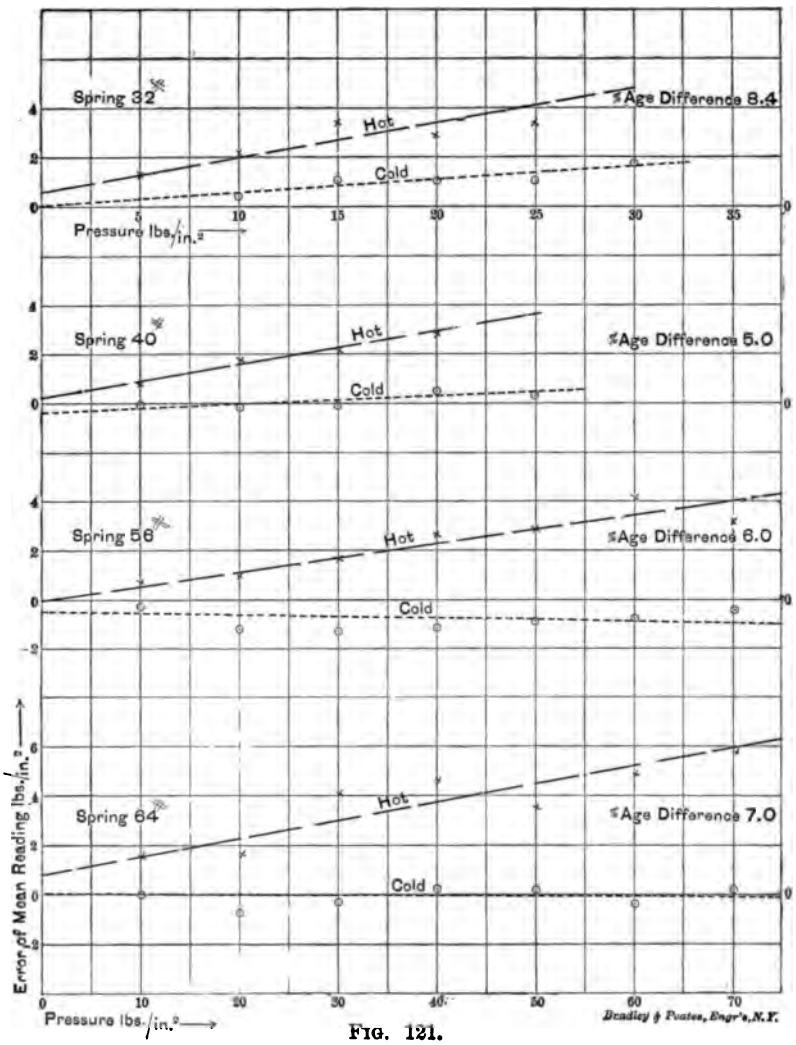


FIG. 121.

INDICATOR I.

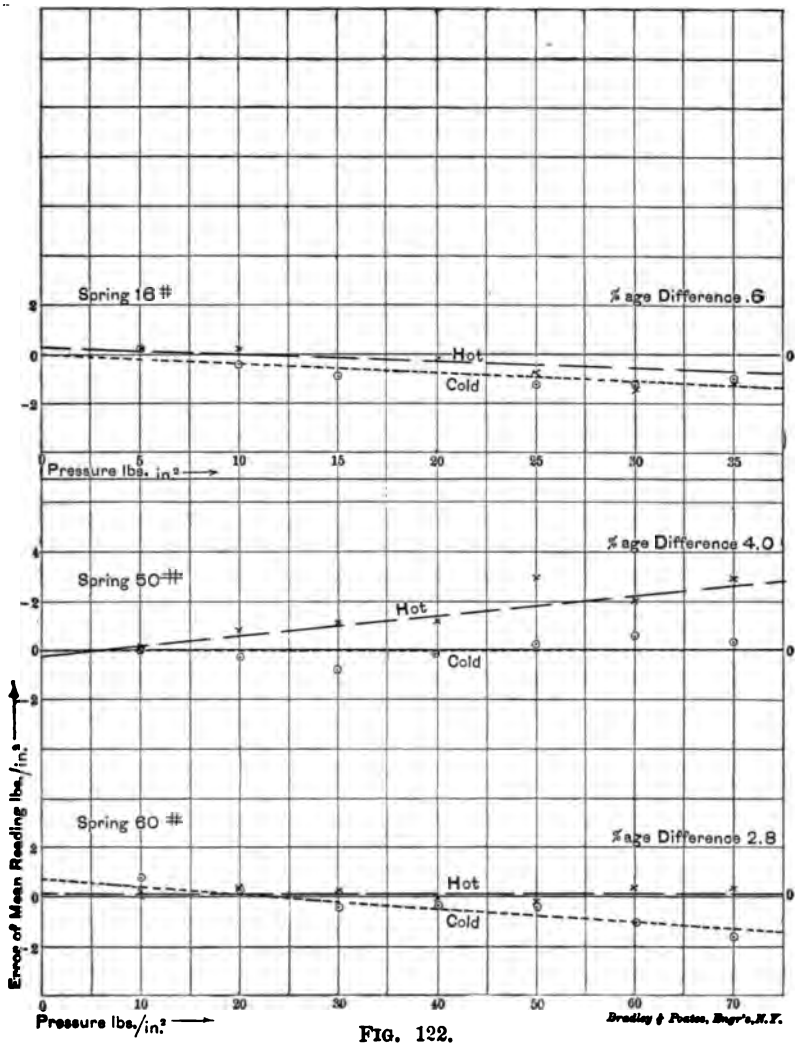


FIG. 122.

INDICATOR G.

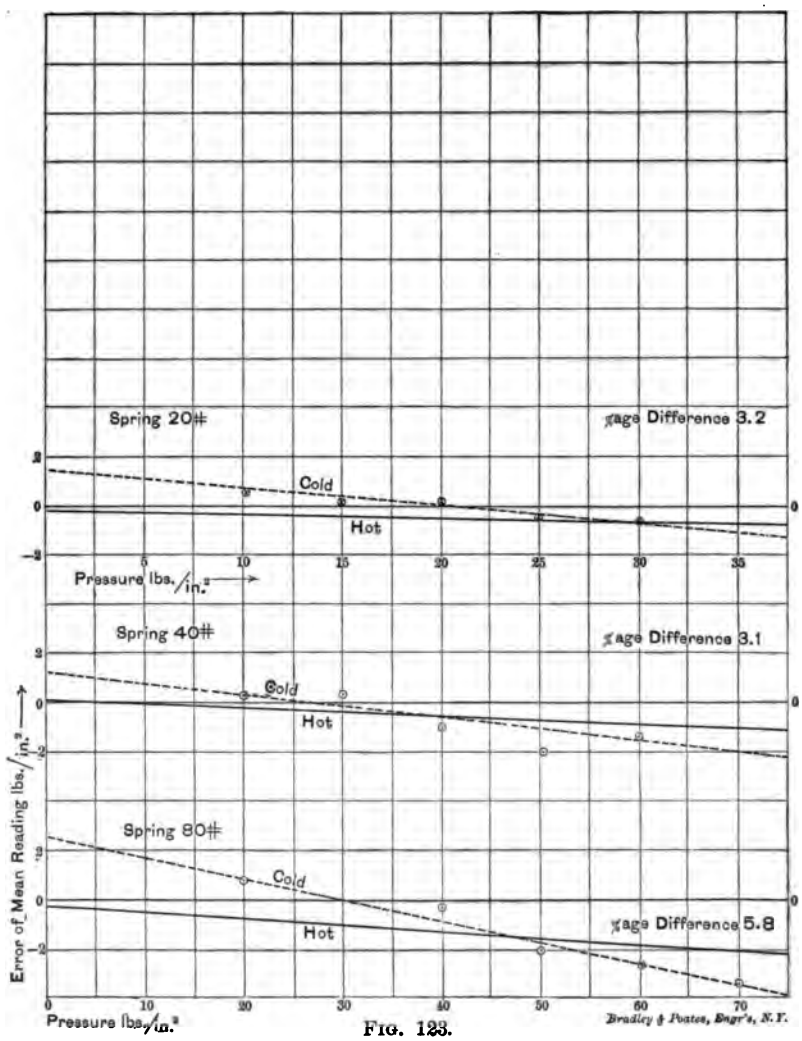


FIG. 123.

INDICATOR E.

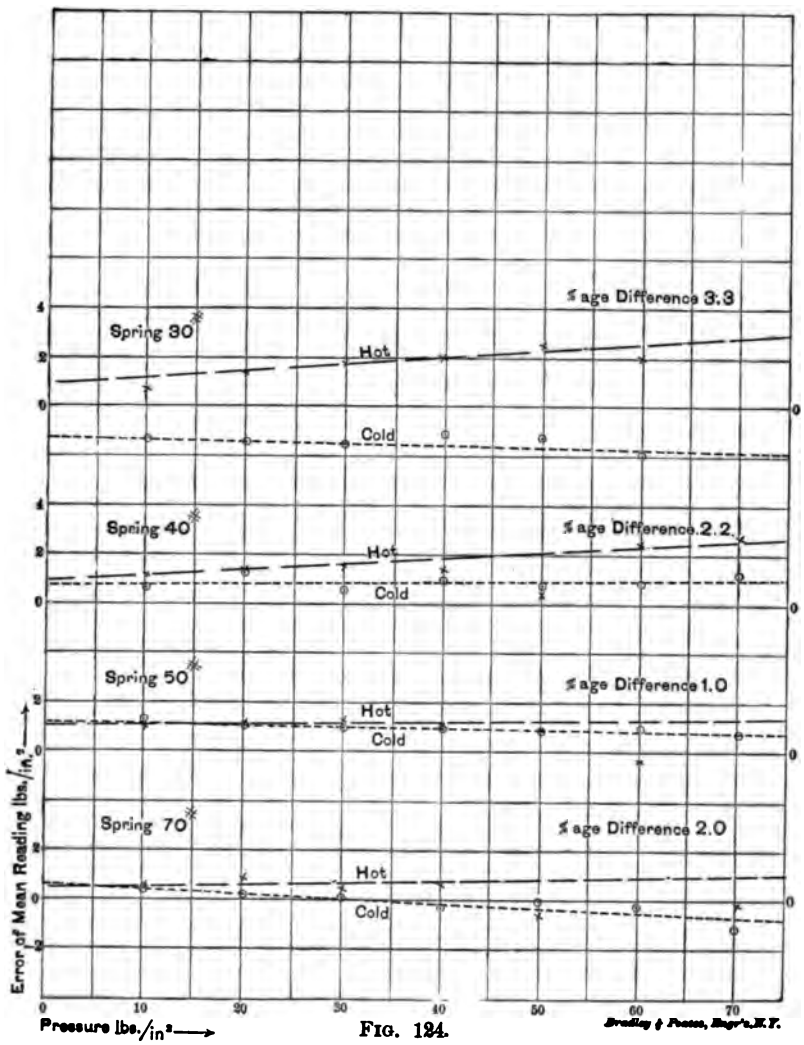


FIG. 124.

INDICATOR E.

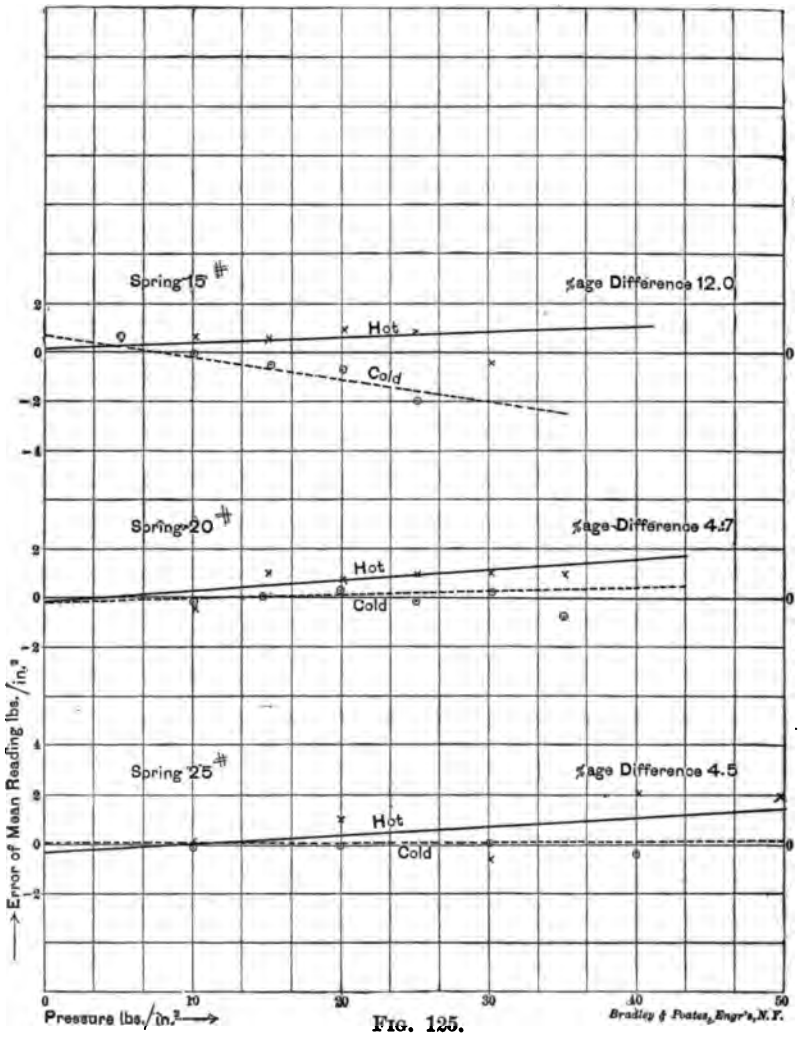


FIG. 125.

INDICATOR C.

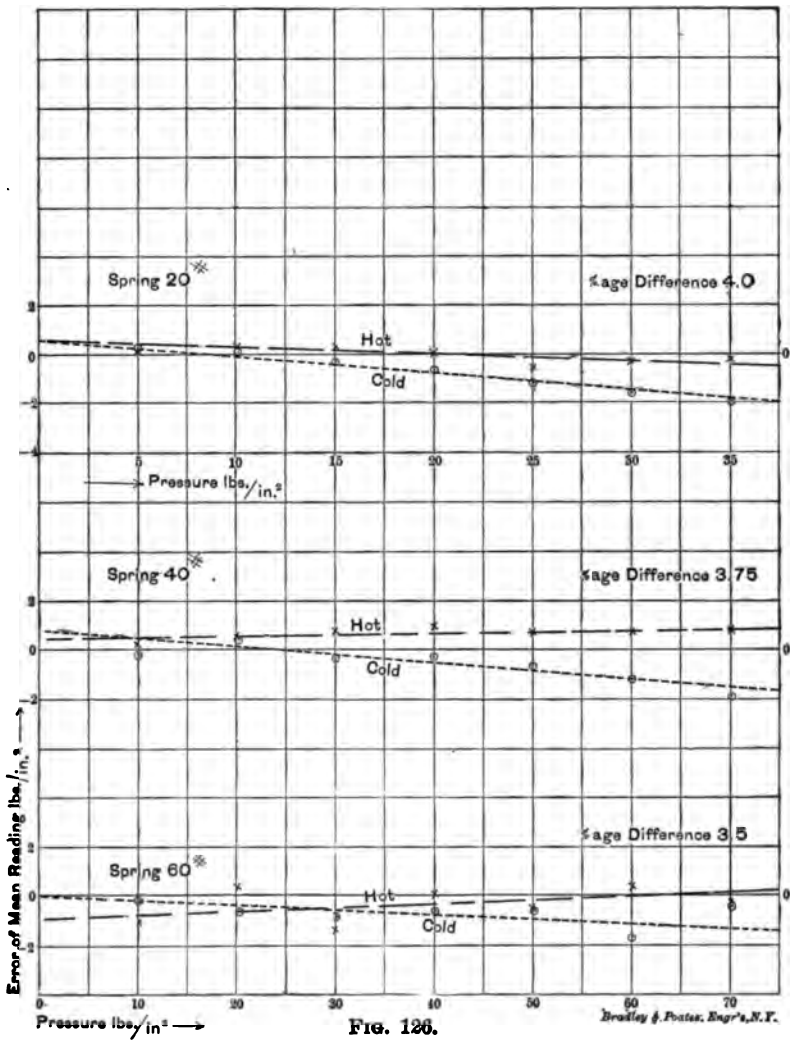


FIG. 126.

RESULTS OF TEST OF SPRINGS FOR INDICATOR D.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	Cold Test.					Hot Test.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
24	9	30	10	10.1	8.9	9.3	9.1	.....	−1.0	11.7	12.3	12.0	.....	1.9	.....	
			15	15.1	13.9	14.3	14.2	.....	− .9	.....	.....	.....	.....	.....	.....	
			20	20.1	17.7	19.5	18.6	.....	−1.5	21.2	22.2	21.7	.....	1.6	.....	
			25	25.2	22.2	24.0	23.1	1.5	−2.1	25.4	27.7	26.6	1.0	1.4	4.2	
			30	30.2	26.5	29.2	27.9	.....	−2.3	30.0	32.6	31.3	.....	1.1	.....	
			35	35.2	30.8	34.1	32.5	.....	−2.7	35.1	37.4	36.3	.....	1.1	.....	
			40	40.3	34.9	39.1	37.0	.....	−3.3	39.6	42.0	40.8	.....	.5	.....	
			45	45.3	39.1	43.6	41.4	.....	−3.9	44.4	.....	.....	.....	.....	.....	
			50	50.4	43.6	....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
21	11	48	10	10.1	10.5	10.7	10.6	.....	.5	.....	.....	.....	.....	.....	.....	
			15	15.1	.....	.....	.....	.....	.....	15.8	16.0	15.9	.....	.8	.....	
			20	20.1	20.3	20.6	20.5	.....	.4	.....	.....	.....	.....	.....	.....	
			30	30.2	30.3	29.1	29.7	.....	− .5	31.0	31.2	31.1	.....	.9	.....	
			40	40.3	40.5	41.4	41.0	.2	.7	39.0	41.0	40.0	.5	− .3	4.6	
			50	50.4	49.8	50.8	50.3	.....	− .1	52.8	53.2	53.0	.....	2.6	.....	
			60	60.4	60.0	60.2	60.1	.....	− .3	61.5	61.8	61.7	.....	1.3	.....	
			70	70.5	69.5	69.9	69.7	.....	− .8	71.6	74.2	72.9	.....	2.4	.....	
			75	75.5	.....	.....	.....	.....	.....	76.5	.....	.....	.....	.....	.....	
23	10	56	10	10.1	9.0	9.2	9.1	.....	−1.0	.....	.....	.....	.....	.....	.....	
			15	15.1	.....	.....	.....	.....	.....	13.5	15.4	14.5	.....	− .6	.....	
			20	20.1	17.7	18.5	18.1	.....	−2.0	.....	.....	.....	.....	.....	.....	
			30	30.2	27.2	28.2	27.7	.....	−2.5	27.8	30.3	29.1	.....	−1.1	.....	
			40	40.3	35.4	36.4	35.9	.5	−4.4	37.0	39.5	38.3	1.1	−2.0	2.8	
			50	50.4	44.9	46.0	45.5	.....	−4.9	47.0	48.0	47.5	.....	−2.9	.....	
			60	60.4	54.0	55.5	54.8	.....	−5.6	56.0	58.0	57.0	.....	−3.4	.....	
			70	70.5	63.5	.....	.....	.....	.....	64.5	68.0	66.3	.....	−4.2	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR D.—*Continued.*

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	COLD TEST.					HOT TEST.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
22	12	64	10	10.1	9.5	10.2	9.9	.....	− .2	.....	.....	.....	.....	.....	.....	
			20	20.1	19.3	19.6	19.5	.....	− .6	21.8	23.0	22.4	.....	1.3	.....	
			30	30.2	30.5	30.7	30.6	.....	.4	32.9	34.1	33.5	.....	3.3	.....	
			40	40.3	41.3	40.6	41.0	.125	.7	42.9	44.1	43.5	.5	3.2	3.4	
			50	50.4	51.0	52.0	51.5	.....	1.1	53.4	54.2	53.8	.....	3.4	.....	
			60	60.4	60.8	60.8	60.8	.....	.4	63.4	64.2	63.8	.....	3.4	.....	
			70	70.5	71.2	.....	.....	.....	.....	73.9	74.5	74.2	.....	3.7	.....	
			80	80.6	.....	.....	.....	.....	.....	84.5	.....	.....	.....	.....	.....	
36	13	80	10	10.1	7.6	9.8	8.7	.....	−1.4	.....	.....	.....	.....	.....	.....	
			20	20.1	16.6	19.2	17.9	.....	−2.2	20.0	22.0	21.0	.....	.9	.....	
			30	30.2	25.8	29.6	27.7	.....	−2.5	30.0	32.8	31.4	.....	1.2	.....	
			40	40.3	35.2	40.8	38.0	.45	−2.3	39.2	43.2	41.2	1.8	.9	3.0	
			50	50.4	45.4	50.2	47.8	.....	−2.6	49.2	52.8	51.0	.....	.6	.....	
			60	60.4	54.2	60.6	57.4	.....	−3.0	58.8	63.4	61.1	.....	.7	.....	
			70	70.5	64.0	.....	.....	.....	.....	68.8	73.4	71.1	.....	.6	.....	
			80	80.6	.....	.....	.....	.....	.....	79.2	82.4	80.8	.....	.2	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR G

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	COLD TEST.					HOT TEST.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.		Average friction, (from diag.).	Scale reading. lbs.	Equivalent pressure, lbs. per sq. in.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.			Up.	Down.					
15	2	20	10	10.1	10.0	11.3	10.7	.....	.6	9.0	.....	.....	9.2	9.3	.....	
			15	15.1	14.4	15.9	15.2	.....	.1	12.5	.....	.....	12.6	12.7	.....	
			20	20.1	19.3	21.1	20.2	.9	.1	18.1	.....	.1	18.1	18.2	3.2	
			25	25.2	23.5	25.9	24.7	.....	—.5	21.4	.....	.....	21.7	21.8	.....	
			30	30.2	28.2	30.9	29.6	.....	—.6	28.5	.....	.....	28.7	28.9	.....	
			35	35.2	33.1	.....	.....	.....	.....	31.7	.....	.....	31.9	32.1	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	9.3	.....	9.1	9.2	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	12.9	.....	12.9	13.1	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	17.3	.....	17.5	17.6	.....	
14	4	40	20	20.1	19.5	21.2	20.4	.....	.3	10.8	.....	.....	11.5	11.6	.....	
			30	30.2	29.7	31.3	30.5	.....	.3	19.3	.....	.....	19.1	19.2	.....	
			40	40.3	37.8	40.6	39.2	1.2	—1.1	30.0	.....	1.0	31.0	31.2	3.1	
			50	50.4	47.5	49.3	48.4	.....	—2.0	39.3	.....	.....	41.0	41.3	.....	
			60	60.4	56.8	61.2	59.0	.....	—1.4	49.4	.....	.....	51.2	51.4	.....	
			70	70.5	65.3	.....	.....	.....	.....	59.5	.....	.....	61.6	62.0	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	49.7	.....	48.7	49.1	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	32.1	.....	31.0	31.2	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	19.7	.....	19.8	19.9	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	7.5	.....	7.1	7.1	.....	
16	5	80	20	20.1	19.5	22.2	20.9	.....	.8	32.4	.....	.....	31.0	31.2	.....	
			40	40.3	37.8	42.2	40.0	.....	— .3	52.0	.....	.....	50.0	50.4	.....	
			50	50.4	46.8	50.0	48.4	2.1	—2.0	59.2	.....	—2.0	59.3	59.7	5.8	
			60	60.4	55.0	60.6	57.8	.....	—2.6	69.2	.....	.....	70.1	70.6	.....	
			70	70.5	63.4	70.8	67.1	.....	—3.4	81.0	.....	.....	83.2	83.8	.....	
			77.4	78.0	73.2	.....	.....	.....	.....	91.2	.....	.....	91.5	92.2	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	77.0	.....	80.4	81.0	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	67.0	.....	70.0	70.5	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	57.0	.....	59.8	60.2	.....	
			.....	.....	.....	.....	.....	.....	.....	.....	31.8	.....	37.5	37.8	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR E.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	Cold Test.					Hot Test.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
45	55	30	10	10.1	7.7	9.8	8.8	.....	−1.3	9.5	12.0	10.8	.....	.7	.....	
			20	20.1	17.7	19.7	18.7	.....	−1.4	19.9	22.9	21.4	.....	1.3	.....	
			30	30.2	27.2	29.9	28.6	.....	−1.6	30.4	33.6	32.0	.....	1.8	.....	
			40	40.3	37.8	40.5	39.2	1.1	−1.1	40.8	43.8	42.3	1.6	2.0	3.3	
			50	50.4	47.7	50.5	49.1	.....	−1.3	50.4	55.4	52.9	.....	2.5	.....	
			60	60.4	57.9	58.9	58.4	.....	−2.0	60.8	63.9	62.4	.....	2.0	.....	
46	61	40	10	10.1	9.6	11.8	10.7	.....	.6	11.2	11.3	11.3	.....	1.2	.....	
			20	20.1	20.7	22.2	21.5	.....	1.4	20.8	22.2	21.5	.....	1.4	.....	
			30	30.2	29.5	32.2	30.9	.....	.7	30.7	32.8	31.8	.....	1.5	.....	
			40	40.3	39.8	42.8	41.3	1.1	1.0	40.6	42.8	41.7	.7	1.4	2.2	
			50	50.4	49.5	52.7	51.1	.....	.7	51.0	50.4	50.7	.....	.3	.....	
			60	60.4	59.9	62.6	61.3	.....	.9	61.0	64.6	62.8	.....	2.4	.....	
			70	70.5	70.6	72.7	71.7	.....	1.2	71.8	74.6	73.2	.....	2.7	.....	
			80	80.6	81.2	81.2	81.2	.....	.6	82.8	83.2	83.0	.....	2.4	.....	
47	56	50	10	10.1	9.8	12.6	11.2	.....	1.1	10.2	11.8	11.0	.....	.9	.....	
			20	20.1	19.7	22.5	21.1	.....	1.0	20.0	22.4	21.2	.....	1.1	.....	
			30	30.2	29.5	32.8	31.2	.....	1.0	30.3	32.6	31.5	.....	1.3	.....	
			40	40.3	39.6	42.8	41.2	1.4	.9	40.1	42.6	41.4	1.7	1.1	1.0	
			50	50.4	49.4	52.9	51.2	.....	.8	50.0	52.5	51.3	.....	.9	.....	
			60	60.4	59.4	63.2	61.3	.....	.9	60.9	60.9	60.1	.....	−.3	.....	
			70	70.5	69.3	73.3	71.3	.....	.8	70.8	70.8	70.2	.....	−.3	.....	
			80	80.6	79.5	79.5	79.5	.....	−1.1	.....	.....	.....	.....	.....	.....	
48	57	70	10	10.1	9.5	11.4	10.5	.....	.4	9.1	12.3	10.7	.....	.6	.....	
			20	20.1	19.1	21.4	20.3	.....	.2	19.4	22.1	20.8	.....	.7	.....	
			30	30.2	29.1	31.2	30.3	.....	.1	29.2	32.2	30.7	.....	.5	.....	
			40	40.3	39.2	40.8	40.0	1.1	− .3	39.4	42.4	40.9	1.5	.6	2.0	
			50	50.4	48.7	51.9	50.3	.....	− .1	47.3	52.2	49.8	.....	− .6	.....	
			60	60.4	58.8	61.3	60.1	.....	− .3	60.6	62.1	61.4	.....	1.0	.....	
			70	70.5	67.9	70.7	69.3	.....	−1.2	68.3	71.4	69.9	.....	− .6	.....	
			80	80.6	78.4	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR E.—*Continued.*

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	COLD TEST.					HOT TEST.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
40	54	15	5	5.	3.4	7.7	5.6	.....	.6	.....	.....	.....	.....	.....	.....	
			10	10.1	7.0	13.1	10.1	.....	.0	9.6	11.3	10.5	.....	.4	.....	
			15	15.1	11.4	17.5	14.5	2.6	− .6	14.6	16.7	15.7	.9	.6	12.0	
			20	20.1	16.4	22.4	19.4	.....	− .7	19.7	21.4	20.6	.....	.5	.....	
			25	25.2	21.5	24.9	23.2	.....	−2.0	25.2	26.9	26.1	.....	.9	.....	
			30	30.2	26.4	27.3	26.9	.....	−3.3	29.1	30.5	29.8	.....	− .4	.....	
51	52	20	10	10.1	7.5	12.0	9.8	.....	− .3	9.4	10.0	9.7	.....	− .4	.....	
			15	15.1	12.3	18.0	15.2	.....	.1	15.4	16.8	16.1	.....	1.0	.....	
			20	20.1	17.7	23.0	20.4	.....	.3	19.4	22.2	20.8	.....	.7	.....	
			25	25.2	22.6	27.5	25.1	2.5	− .1	24.8	27.6	26.2	1.1	1.0	4.7	
			30	30.2	27.5	33.4	30.5	.....	.3	29.8	32.8	31.3	.....	1.1	.....	
			35	35.2	32.7	36.4	34.6	.....	− .6	34.4	36.9	35.7	.....	.5	.....	
			40	40.3	38.2	38.7	38.5	.....	−1.8	40.0	40.3	40.2	.....	− .1	.....	
50	53	25	10	10.1	9.9	12.7	10.0	.....	− .1	9.6	10.5	10.1	.....	.0	.....	
			20	20.1	19.9	23.3	20.1	.....	.0	19.5	22.8	21.2	.....	1.1	.....	
			30	30.2	29.8	33.3	30.4	2.8	.2	30.0	29.3	29.7	1.2	− .5	4.5	
			40	40.3	39.7	42.5	40.0	.....	− .3	39.3	45.5	42.4	.....	2.1	.....	
			50	50.4	49.7	49.0	48.0	.....	−2.4	49.7	51.5	50.6	.....	.2	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR H.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	COLD TEST.					HOT TEST.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
35	58	20	5	5.	5.3	4.3	4.8	.....	−.2	6.1	5.2	5.6	.....	.6	.....	Some lost motion, friction apparently negative.
			10	10.1	10.3	10.1	10.2	.....	.1	12.4	10.6	11.5	.....	1.4	.....	
			15	15.1	15.6	15.2	15.4	−.4	.3	18.0	16.0	17.0	.....	1.9	.....	
			20	20.1	20.7	19.4	20.1	.....	0	22.9	21.3	22.1	−.6	2.0	2.9	
			25	25.2	25.9	25.5	25.7	.....	.2	27.4	26.7	27.1	.....	1.9	.....	
			30	30.2	31.2	30.3	30.8	.....	.6	32.1	31.9	32.0	.....	1.8	.....	
			35	35.2	36.6	.....	.....	.....	.....	38.5	37.5	38.0	.....	2.8	.....	
34	59	40	10	10.1	10.2	9.9	10.1	.....	0	11.2	10.5	10.9	.....	.8	.....	
			20	20.1	20.2	20.2	20.2	.....	.1	21.2	20.9	21.1	.....	1.0	.....	
			30	30.2	30.5	29.6	30.1	.....	−.1	32.4	32.0	32.2	.....	2.0	.....	
			40	40.3	40.8	41.0	40.9	.1	.6	43.1	43.0	43.1	−.1	2.8	4.7	
			50	50.4	51.0	52.0	51.5	.....	1.1	54.1	54.1	54.1	.....	3.7	.....	
			60	60.4	61.3	62.2	61.8	.....	1.4	64.6	64.6	64.6	.....	4.2	.....	
			70	70.5	71.5	72.6	72.1	.....	1.6	75.4	75.4	75.4	.....	4.9	.....	
			80	80.6	82.1	82.2	82.2	.....	1.6	.....	.....	.....	.....	.....	.....	
33	60	60	10	10.1	9.8	9.3	9.6	.....	−.5	11.7	9.6	10.7	.....	.6	.....	Some lost motion as above.
			20	20.1	19.8	19.4	19.6	.....	−.5	21.0	20.0	20.5	.....	.4	.....	
			30	30.2	30.3	29.7	30.0	.....	−.2	31.4	31.1	31.3	.....	1.1	.....	
			40	40.3	40.1	39.5	39.8	−.4	−.5	41.7	41.3	41.5	.4	1.2	3.0	
			50	50.4	50.6	49.2	49.9	.....	−.5	51.9	51.0	51.5	.....	1.1	.....	
			60	60.4	60.9	59.3	60.1	.....	−.3	63.0	61.7	62.4	.....	2.0	.....	
			70	70.5	70.4	70.1	70.3	.....	−.2	72.8	72.8	72.8	.....	2.3	.....	
			80	80.6	81.3	79.5	80.4	.....	−.2	.....	.....	.....	.....	.....	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR F.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	Cold Test.					Hot Test.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
25	32	16	5	5.	5.0	4.8	4.9	.....	− .1	5.2	4.3	4.8	.....	− .2	.....	
			10	10.1	9.0	9.2	9.1	.....	−1.0	9.9	9.5	9.7	.....	− .4	.....	
			15	15.1	13.6	14.0	13.8	.2	−1.3	14.6	14.0	14.3	−.15	− .8	5.3	
			20	20.1	18.0	19.0	18.5	.....	−1.6	19.7	19.0	19.4	.....	− .7	.....	
			25	25.2	22.8	23.4	23.1	.....	−2.1	23.7	24.3	24.0	.....	−1.2	.....	
			30	30.2	27.3	27.9	27.6	.....	−2.6	28.4	28.7	28.6	.....	−1.6	.....	
26	31	40	10	10.1	9.8	10.5	10.2	.....	.1	9.8	10.0	9.9	.....	− .2	.....	
			20	20.1	19.3	20.3	19.8	.....	− .3	19.9	20.4	20.2	.....	.1	.....	
			30	30.2	28.4	30.0	29.2	.....	−1.0	29.7	30.4	30.1	.....	− .1	.....	
			40	40.3	38.4	40.2	39.3	.7	−1.0	39.5	40.8	40.2	.5	− .1	3.4	
			50	50.4	48.2	50.3	49.3	.....	−1.1	50.0	51.2	50.6	.....	.2	.....	
			60	60.4	58.2	60.4	59.3	.....	−1.1	60.1	61.6	60.9	.....	.5	.....	
			70	70.5	68.4	70.2	69.3	.....	−1.2	70.3	72.2	71.3	.....	.8	.....	
			80	80.6	78.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
27	30	50	10	10.1	9.5	9.5	9.5	.....	− .6	10.6	10.6	10.6	.....	.5	.....	
			20	20.1	18.2	18.2	18.2	.....	−1.9	20.9	20.9	20.9	.....	.8	.....	
			30	30.2	28.5	29.0	28.8	.....	−1.4	30.8	30.8	30.8	.....	.6	.....	
			40	40.3	38.2	38.6	38.4	.....	−1.9	40.6	40.6	40.6	.....	.3	.....	
			50	50.4	48.0	48.0	48.0	.05	−2.4	50.6	50.6	50.6	0	.2	4.3	
			60	60.4	58.3	58.3	58.3	.....	−2.1	60.8	60.8	60.8	.....	.4	.....	
			70	70.5	68.0	66.9	67.5	.....	−3.0	70.8	70.8	70.8	.....	.3	.....	
			80	80.6	77.2	77.2	77.2	.....	−3.4	81.0	81.0	81.0	.....	.4	.....	
28	29	60	10	10.1	9.6	9.0	9.3	.....	− .8	9.8	10.2	10.0	.....	− .1	.....	
			20	20.1	19.4	18.6	19.0	.....	−1.1	19.7	20.4	20.1	.....	0	.....	
			30	30.2	28.2	28.5	28.4	.....	−1.8	29.0	30.0	29.5	.....	− .7	.....	
			40	40.3	37.4	38.1	37.8	.2	−2.5	38.7	39.3	39.0	.85	−1.3	4.5	
			50	50.4	46.8	48.2	47.5	.....	−2.9	48.2	50.8	49.5	.....	− .9	.....	
			60	60.4	55.8	55.8	55.8	.....	−4.6	58.2	60.9	59.6	.....	− .8	.....	
			70	70.5	65.1	66.9	66.0	.....	−4.5	67.5	70.2	68.9	.....	−1.1	.....	
			80	80.6	.....	.....	.....	.....	.....	78.0	80.7	79.4	.....	−1.2	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR B.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure, lbs. per sq. in.	Cold Test.					Hot Test.					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
38	44	32	10	10.1	9.8	11.2	10.5	.....	.4	10.7	12.0	11.4	.....	1.3	.....	
			15	15.1	15.4	16.8	16.1	.....	1.0	16.9	17.4	17.2	.....	2.1	.....	
			20	20.1	19.2	22.9	21.1	1.6	1.0	23.0	23.9	23.5	.5	3.4	8.4	
			25	25.2	24.1	28.3	26.2	.....	1.0	27.3	28.9	28.1	.....	2.9	.....	
			30	30.2	29.2	34.5	31.9	.....	1.7	33.0	34.0	33.5	.....	3.3	.....	
			35	35.2	34.3	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
37	43	40	10	10.1	9.5	10.4	10.0	.....	− .1	10.2	11.6	10.9	.....	.8	.....	
			20	20.1	18.6	21.2	19.9	.....	− .2	20.5	23.0	21.8	.....	1.7	.....	
			30	30.2	28.4	31.8	30.1	1.5	− .1	31.2	33.4	32.3	1.0	2.1	5.0	
			40	40.3	39.0	42.6	40.8	.....	.5	42.1	44.0	43.1	.....	2.8	.....	
			50	50.4	48.4	53.0	50.7	.....	.3	.....	.....	.....	.....	.....	.....	
39	42	56	10	10.1	8.5	11.1	9.8	.....	− .3	9.3	12.2	10.8	.....	.7	.....	
			20	20.1	16.4	21.3	18.9	.....	−1.2	19.8	22.2	21.0	.....	.9	.....	
			30	30.2	25.0	32.8	28.9	.....	−1.3	30.2	33.4	31.8	.....	1.6	.....	
			40	40.3	35.6	42.6	39.1	3.4	−1.2	40.8	45.0	42.9	1.4	2.6	6.0	
			50	50.4	45.5	53.4	49.5	.....	− .9	52.0	54.4	53.2	.....	2.8	.....	
			60	60.4	55.0	64.2	59.6	.....	− .8	63.0	66.0	64.5	.....	4.1	.....	
			70	70.5	66.2	74.0	70.1	.....	− .4	72.8	74.4	73.6	.....	3.1	.....	
			80	80.6	76.0	.....	.....	.....	.....	.....	.....	.....	.....	.....	.....	
40	41	64	10	10.1	8.9	11.2	10.1	.....	0	10.5	12.6	11.6	.....	1.5	.....	
			20	20.1	17.8	21.0	19.4	.....	− .7	20.0	23.4	21.7	.....	1.6	.....	
			30	30.2	27.2	32.5	29.9	.....	− .3	34.0	34.5	34.3	.....	4.1	.....	
			40	40.3	37.0	44.0	40.5	3.3	.2	43.0	45.8	44.9	1.9	4.6	7.0	
			50	50.4	46.0	55.0	50.5	.....	.1	51.8	56.0	53.9	.....	3.5	.....	
			60	60.4	54.5	65.5	60.0	.....	− .4	62.5	68.0	65.3	.....	4.9	.....	
			70	70.5	66.2	75.0	70.6	.....	.1	72.0	80.5	76.3	.....	5.8	.....	
			80	80.6	74.5	.....	.....	.....	.....	83.0	.....	.....	.....	.....	.....	

RESULTS OF TEST OF SPRINGS FOR INDICATOR I.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure lbs. per sq. in.	Cold Test.					Hot Test					Av. % difference hot and cold.	Remarks.
					Height of Ordinates.			Average friction. lbs.	Error of mean reading.	Height of Ordinates.			Average friction. lbs.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.			Up.	Down.	Mean.				
65	70	16	5	5.0	5.0	5.6	5.3	.....	.3	4.6	6.0	5.3	.....	.3	.....	
			10	10.1	9.6	10.0	9.8	.....	− .3	9.1	11.1	10.1	.....	.0	.....	
			15	15.1	14.0	14.6	14.3	.....	− .8	14.0	16.2	15.1	.....	.0	.....	
			20	20.1	.....	19.3	.....	.26	.....	18.2	21.7	20.0	1.6	− .1	.6	
			25	25.2	23.8	24.2	24.0	.....	−1.2	22.6	26.3	24.5	.....	− .7	.....	
			30	30.2	28.7	29.2	29.0	.....	−1.2	26.9	30.8	28.9	.....	−1.3	.....	
			35	35.2	33.9	34.7	34.3	.....	− .9	31.2	36.9	34.1	.....	−1.1	.....	
66	69	50	10	10.1	9.4	10.7	10.1	.....	0	8.5	11.9	10.2	.....	.1	.....	
			20	20.1	19.2	20.6	19.9	.....	− .2	19.4	22.4	20.9	.....	.8	.....	
			30	30.2	29.4	29.4	29.4	.....	− .8	29.9	32.6	31.3	.....	1.1	.....	
			40	40.3	39.2	41.2	40.2	.55	− .1	39.5	43.5	41.5	1.8	1.2	4.0	
			50	50.4	49.8	51.4	50.6	.....	.2	52.0	54.8	53.4	.....	3.0	.....	
			60	60.4	60.5	61.4	61.0	.....	.6	60.0	64.7	62.4	.....	2.0	.....	
			70	70.5	70.3	71.3	70.8	.....	.3	70.9	75.8	73.4	.....	2.9	.....	
			80	80.6	80.1	80.9	80.5	.....	− .1	82.1	86.3	84.2	.....	3.6	.....	
67	68	60	10	10.1	9.5	12.0	10.8	.....	.7	8.5	12.0	10.3	.....	.2	.....	
			20	20.1	19.1	21.5	20.3	.....	.2	18.5	22.2	20.4	.....	.3	.....	
			30	30.2	28.5	31.0	29.8	.....	− .4	28.7	32.2	30.5	.....	.3	.....	
			40	40.3	39.7	40.4	40.1	.50	− .2	38.0	42.5	40.3	2.4	.0	2.8	
			50	50.4	50.0	50.2	50.1	.....	− .3	47.7	52.9	50.3	.....	− .1	.....	
			60	60.4	59.4	59.4	59.4	.....	−1.0	58.5	63.0	60.8	.....	.4	.....	
			70	70.5	69.1	69.1	69.1	.....	−1.6	67.5	74.0	70.8	.....	.3	.....	
			80	80.6	78.5	78.5	78.5	.....	−2.1	77.0	84.2	80.6	.....	.0	.....	

RESULTS OF TEST OF SPRINGS OF INDICATOR C.

No. of Card.		Scale of spring.	Scale reading.	Equivalent pressure per lb. per sq. in.	COLD TEST.					HOT TEST.					Av. % difference of readings hot and cold.	Remarks.
					Height of Ordinates.			Average friction.	Error of mean reading.	Height of Ordinates.			Average friction.	Error of mean reading.		
Cold.	Hot.		lbs.		Up.	Down.	Mean.	lbs.		Up.	Down.	Mean.	lbs.			
71	72	20	5	5.0	5.1	5.0	5.1	.....	.1	5.2	5.0	5.1	.....	.1	.....	
			10	10.1	10.6	9.9	10.3	.....	.2	10.5	10.2	10.4	.....	.3	.....	
			15	15.1	15.1	14.3	14.7	.....	− .4	15.5	15.2	15.4	.....	.3	.....	
			20	20.1	19.4	19.4	19.4	.....	− .7	20.3	19.9	20.1	.....	.0	.....	
			25	25.2	24.0	24.0	24.0	.25	−1.2	25.0	24.2	24.6	.06	− .6	4.0	
			30	30.2	28.5	28.8	28.7	.....	−1.5	29.4	30.3	29.9	.....	− .3	.....	
			35	35.2	32.0	34.5	33.3	.....	−1.9	34.2	35.8	35.0	.....	− .2	.....	
			40	40.3	36.6	39.2	37.9	.....	−2.4	39.0	39.5	39.3	.....	−1.0	.....	
73	74	40	10	10.1	9.9	9.9	9.9	.....	− .2	10.6	10.0	10.3	.....	.2	.....	
			20	20.1	20.4	20.4	20.4	.....	.3	20.8	20.4	20.6	.....	.5	.....	
			30	30.2	29.8	29.8	29.8	.....	− .4	31.2	30.6	30.9	.....	.7	.....	
			40	40.3	40.0	40.0	40.0	.1	− .3	41.4	40.8	41.1	−.3	.8	3.75	
			50	50.4	49.8	49.6	49.7	.....	− .7	51.4	50.8	51.1	.....	.7	.....	
			60	60.4	59.2	59.2	59.2	.....	−1.2	61.4	60.8	61.1	.....	.7	.....	
			70	70.5	67.8	69.4	68.6	.....	−1.9	71.6	70.8	71.2	.....	.7	.....	
75	76	60	10	10.1	9.9	9.9	9.9	.....	− .2	8.5	9.5	9.0	.....	−1.1	.....	
			20	20.1	19.4	19.4	19.4	.....	− .7	20.4	20.4	20.4	.....	.3	.....	
			30	30.2	28.9	29.8	29.4	.....	− .8	30.2	27.4	28.8	.....	−1.4	.....	
			40	40.3	39.6	39.6	39.6	.....	− .7	40.9	39.7	40.3	.....	.0	.....	
			50	50.4	49.4	50.1	49.8	−.06	− .6	50.2	49.5	49.9	−.7	− .5	3.5	
			60	60.4	59.2	58.0	58.6	.....	−1.8	62.0	59.6	60.8	.....	.4	.....	
			70	70.5	70.0	70.0	70.0	.....	− .5	71.2	69.3	70.3	.....	− .2	.....	
			80	80.5	78.9	77.5	78.2	.....	−2.3	80.7	77.2	79.0	.....	−1.5	.....	

DISCUSSION.

Mr. Gus. C. Henning.—I would like to ask Professor Carpenter whether he found that the modulus of elasticity of those springs would vary 3% under a temperature change of about 160°, taking the normal temperature at which those springs were tested at 60°. What is the temperature you state?

Professor Carpenter.—About 60° to 220° Fahr. Both the results of calculation and experiment agree.

Mr. Henning.—Was that the temperature of the steam?

Professor Carpenter.—The temperature of the steam was estimated at 220° Fahr.; it may have been 8° or 10° less, or even that much higher.

Mr. Henning.—Did you test those springs and find there was a difference in the modulus of elasticity of 3%?

Professor Carpenter.—The test only covered the ground indicated in the paper.

Mr. Henning.—That 3% would make 900,000 lbs. in the modulus of elasticity, and such a thing has never been found before. I understand you to say that the errors of the springs were checked off theoretically by calculation and substitution of the elastic formula, assuming the modulus of elasticity, and that that would give you an error of up to 6.6%.

Professor Carpenter.—The theoretical calculation is given on page 467. The modulus of elasticity is diminished by 1 minus the fraction .00019. The modulus of elasticity referred to here is not that of the steel wire, but that of the coiled spring. Now, it is quite evident that Mr. Henning is talking about the modulus of the steel, which is about 30,000,000. I refer to the modulus of the coiled spring, which is about 50,000. The computation is from an independent investigation made with coiled springs by Mr. Marks, at Mason College, England. I looked over the investigations made, and considered his experiments as excellent. By substituting the coefficient found in Marks's experiments the results come out very close to what was actually found. It seems to my mind a substantiation of its relative accuracy. It must be very nearly right, anyway.

It may be a matter of interest to learn that our method of testing indicator springs, as described in the paper, is one adopted after trying numerous experiments with various standards.

We first compared the indicator springs, when under steam

pressure, with a mercury column. Our results were not at all satisfactory. The springs, apparently, had irregular errors, not at all compatible with Hooke's law of elasticity; furthermore, we found it exceedingly difficult to make our calibration show the same error for the same spring at different times.

Careful investigation convinced me that wave-like pulsations of the mercury were the principal cause of the irregularity, it being a matter of extreme difficulty to determine the true height of the column. For steam-gauge testing this was a matter of no great importance, as the error was much less than the least reading of the gauge; but for indicator-spring testing the error was sufficient to obscure and make the result irregular.

In some recent experiments, to our cost, we have found that what is usually sold as mercury is an amalgam of some metal with specific gravity 10% or 15% less than pure mercury. It seems quite impossible to buy the pure article, and we have recently resorted to a method of distillation to separate out the mercury from the baser metals.

For these various reasons I believe that some system of comparison against standard weights is not only more convenient, but is more accurate and reliable than the mercury column.

The President.—What was the principal impurity which you found in this mercury?

Professor Carpenter.—I am not certain of the composition of the impurities, but regarding the power of mercury to dissolve metals I might mention one thing. Within a very few days we desired to make a metal which would melt at a low temperature, and we found, by consulting tables of melting points, that if we used 50% of mercury and the remaining parts of tin and lead we would have a metal that melted at 106° Fahr.—at least, that was the written statement. We made the mixture, using as per directions 50% of mercury and the proper amount of tin and lead, and we found when we got our metal together that we had a metal that was just as fluid as the mercury, even at 60° Fahr. It looked just like pure mercury. To get a metal that would solidify we had to reduce the mercury to 25% instead of 50%. From this experiment I know that a very large per cent. of tin and lead will be held in solution by mercury, and I am quite sure that there was some tin, and I think, also, some lead in the commercial mercury that we bought for pure mercury.

Mr. Geo. I. Rockwood.—In testing condensing engines I have frequently used a mercury column in place of the vacuum gauge, using mercury as sold in small quantities from the stores.

I see by what Professor Carpenter has said about the effect of the impurity of the commercial mercury on its specific gravity, that it is just as necessary to know all about the mercury used in the column as it is to be assured of the accuracy of the vacuum gauge.

Mr. Charles T. Porter.—I am much interested in the paper of Professor Carpenter. It certainly can do the makers of indicators no harm to know that their work will be subjected to most thorough and searching tests for accuracy. It may be interesting to the society to be told the method that was employed to obtain accuracy in the first manufacture of the Richards indicator in London just thirty years ago. It was a very simple method. I discarded the use of the mercury column, also the use of any manometer guaranteed by anybody, and employed only pieces of iron, the weights of which had been carefully determined. The first object was to get a cylinder of standard and invariable diameter. Having adopted half of a square inch as the area of the cylinder, we applied to the Whitworth Company at Manchester and obtained from them a mandrel of hardened steel, about two feet in length, ground with great care to the diameter, .79789. On this mandrel brass tubes were drawn, about eighteen inches in length, and when forced off from the mandrel they were perfectly true, polished tubes. They were cut up in lengths, each tube making about six cylinders, and so the permanent area of the cylinder was established.

The next step was to test springs for accuracy, and this was done, aside from the instrument altogether, in the open air on a block of metal set on a bench. A rod was put through a hole in the bench and went up through the block and the spring, with a nut on the top. This rod was connected to a lever, the arms of which represented the pencil movement of the indicator, on an enlarged scale. The weights were hung on the lower end of the rod. The block carried an arm which was made to receive the scales for the different springs. In this simple manner the springs were all tested to absolute accuracy throughout their entire range of action. The process of adjusting them is interesting. The heads of the springs were made with three wings, and a spiral groove was cut in these wings so that they

threaded on to the coiled wire. One of the heads was secured on the spring in the following manner: The wire was soldered into two wings, and the third wing was closed on the wire very firmly, so that the vibrations of the spring could not reach the soldered joints. The other head was screwed on loosely, and adjusted in the testing apparatus, so that the spring was a little strong, and that was then secured in the same manner. Then the spring was put in a lathe and the surface was rubbed with emery cloth as it revolved until the reduction in strength brought it precisely to the point of accuracy, and every spring was tested for accuracy through its entire range. That calibration was always cold. The correction for temperature was made in this simple manner: it was assumed that the temperature to which the spring is exposed in use is always about 212°, the upper end of the cylinder being open to the atmosphere. A spring was first tested cold, then left for a minute or two in boiling water, and tested again, when it was found to be weakened one pound in forty. That was invariable, so far as we could observe, for all sizes of wire. So it was easy, having determined this once, to afterwards make the correction when the springs were tested cold. The spring being weighted with 20 lbs. hung on the rod, the index must come to 39 lbs. Then when in use hot it would mark 40 lbs. correctly. That method was so successful that the indicator soon became a standard of pressure by which manometers themselves were tested, and it was found to be quite reliable and invariable. No attempt was made, perhaps there should have been, to ascertain the effect of use. I do not know but that in a long time the springs may have become weakened by use. But at that time we did not go any further in these tests than that. When they went out they were perfect. The wire of which the springs were made was selected with a great deal of care, and a uniform make of wire was used that seemed to stand perfectly. Some samples did not stand so well. I think we were successful in getting a permanently correct spring.

A practically complete frictionlessness was insisted on. The test for friction was simple but searching. The pencil was pressed upward and let down very gently, and a line was drawn. It was then pressed down and allowed to rise in the same manner, and another line was drawn. The two must coincide exactly, forming the same fine line. This was repeated, on the

atmospheric line, and at other points to the top of the scale, the indicator being under different pressures. The workmanship was such that this test was always fully met. The slight torsion of the spring when being compressed was provided for in the pencil connection, and there was no difference in the indication on the upward and downward movements.

Mr. Charles W. Barnaby.—In regard to what Mr. Porter speaks of with reference to the continued accuracy of the springs in service, I had considerable experience in this line some years ago in the early days of the manufacture of the Thompson indicator at the Buckeye Engine Works, Salem, Ohio. We started out without any experience in the line of indicator and spring manufacture, and supposed when we had adjusted the springs to perfect accuracy when cold that we had done all that was required. In the course of time an order came in for a pair of indicators to be used in making an engine test. I made the springs myself, and as the party was particularly anxious to have them absolutely correct, I took great care in adjusting them to the exact strength. After the test was completed the springs came back to be retested, when to my surprise I found that they were uniformly weak to the extent of about one pound in forty. The springs had lost strength in use, and the question arose as to whether it was due to the steam heat or merely to the mechanical effect of the repeated compressions and extensions which the springs had been subjected to. I consulted Mr. Thompson on the subject, but neither of us thought it probable that so low a temperature as 212° could affect the material in such a manner as to weaken the spring. A device was therefore rigged up to operate the springs mechanically, through a range somewhat in excess of the maximum limit of the indicator piston movement. The device operated at a speed of 1200 compressions a minute, and after the springs had been operated in it for one-half minute they were considerably weakened. After being operated for a minute (1200 compressions) they were weakened a very little more. The maximum reduction of a full pound in forty was secured at or below this point, as upon being further subjected to the operation through a period of one, fifteen and even thirty minutes or more, no further reduction of strength was obtained. Although no reduction was observed beyond the first minute's operation, I adopted two minutes (2400 compressions),

as the time during which springs thereafter manufactured should be operated in the device, wishing to keep well on the safe side, and naturally supposed the full solution of the weakening of the springs in use had been reached.

Sometime after this I had occasion to make another set of springs for the same purpose, and when they came back to be retested after the engine test I had a second surprise, as the springs were again too weak, although not as much so as in the first case. There was only one conclusion left, and that was that the springs were affected by the slight heat they were subjected to in use as well as by the mere mechanical effect due to compression.

Fortunately there was a steam pipe handy, and the device for "jigging" the springs, as we called the operation performed upon them, had been so constructed that the spring being "jigged" was enclosed in a cylinder, so that it was an easy matter to connect a steam-pipe to the cylinder. Vent holes were drilled in the lower part of this cylinder to carry off the water of condensation and to give the steam, only a slight amount of which was turned on, a chance to flow freely through and out of the cylinder, so as to obtain practically atmospheric pressure around the spring. It is obvious that this pressure, and its corresponding temperature of 212°, is what is required to place the spring under the same conditions that are met in its use in the indicator; the spring being placed on the top, or atmospheric side of the indicator piston, which space is in constant communication with the atmosphere through vent holes supplied for the purpose of maintaining a constant atmospheric pressure upon the top of the piston. The under side of the piston would be exposed to a much higher pressure and temperature much of the time, and its temperature would thus be considerably above 212°, and part of the heat would be conducted up into the spring, but only for a short distance, as the envelope of lower temperature steam would very rapidly absorb this conducted heat. It is thus evident that under proper conditions the indicator spring could only very slightly exceed a temperature of 212°.

After putting some springs through the first operation and getting all the reduction that was possible by "jigging" cold, I put them in the device again and gave them a second "jigging" under steam, and found that I got another reduction amounting

to about one-half pound, or between one-third and one-half as much reduction as was obtained by the cold "jigging," as near as I can now remember. Having once established the fact that a further reduction was obtainable under heat, there was no further necessity for putting the springs through two separate operations, so I combined the two, giving the springs made thereafter a single two minutes' "jigging" under steam, the results being the same as the aggregate reduction obtained by putting them through the cold and hot operation separately. According to my present recollection, in the first test of the springs when the brass heads were adjusted and soldered on, I left the springs about two pounds in forty strong, the greater part of which was for the reduction obtained by the "jigging" process, and the balance, a small fraction of one pound, was for the final accurate adjustment of the spring to scale by the application of emery cloth as just explained by Mr. Porter. In addition to this two pounds allowance for permanent reduction, one-half pound was allowed for the temporary reduction of strength of the spring while subject to the 212° temperature in use, the springs being left one-half pound in forty strong in the final adjustment.

The weakening of the springs from the effects of the working strains and temperature was very surprising and unaccountable to me. I could reconcile the results due to the mechanical action more to my satisfaction than I could those due to the temperature, and I cannot yet understand how so low a temperature as 212° should permanently affect the strength of the springs as much as it does. While talking with Mr. Porter yesterday on this subject, he suggested that the steel used in constructing the springs might not have been of a proper grade. The springs were bought coiled and tempered, of regular spring makers. I do not think that any special quality of steel was specified. I remember working on one lot of springs on which I could get very little reduction by the "jigging" and heating process, which would seem to indicate that quality of steel has something to do with it. There was only the one lot, so far as I can now recall, out of a large number of springs received from several manufacturers which did not show about a uniform reduction of almost two pounds in forty.

Mr. Thomas R. Almond.—Mr. Porter said something about the quality of the steel used. It seems to me that it is important

to know how the springs were hardened and how they were tempered. Can Mr. Porter say anything about that?

Mr. Porter.—The springs were coiled on grooved mandrels. After the size of wire and pitch had been carefully determined, they were heated over a brazier and tempered. The work was done very uniformly. For several years these springs were all made by one man, a very skilful workman. He certainly was a treasure. He made these springs by the thousand, and they were always correct. The large number made is accounted for by the fact that most purchasers bought a full set of springs.

Mr. Almond.—I would like to say something in regard to a matter that has come to my notice. I find that steel wire acts very irregularly at times. To-day it cannot be coiled on a certain sized mandrel, while six months hence it can be coiled quite easily. That is due, it would seem, to a re-arrangement of the molecules. The internal structure of the steel seems to change so that it is possible to coil a wire, say a No. 11 wire on a $\frac{5}{16}$ mandrel, after it has been standing six months, when it was not possible to coil it before. I am speaking of tempered steel wire, tempered in oil, not a high grade of steel, but of fair quality. I have had about 500 lbs. of this material set aside for the purpose of being coiled after it has been seasoned. I say seasoned. I suppose that is an expression that is used in this connection, and I think it is pretty well understood that wire will improve or mellow up, by time, and I have not a doubt that the springs you speak of are very materially affected by time as well as other things. Take a spring to-day and harden it and temper it and test it, and it will give you such and such results, while if you leave it for six months, without anything being done to it, it will have changed its conditions. I think that the molecules require time to arrange themselves so as to get into the absolutely normal condition, and time alone will permit of this. I think the question of tempering is of serious moment. I remember eighteen years ago sending some tools to a blacksmith who had an excellent reputation, because I did not dare to harden them myself, and the next morning I was surprised to find them broken. The man could not account for it. He said he never had had such an experience before. But the fact was there. From that day to this I have had no bad luck in hardening, because I have hardened everything that was important myself. The secret with me, so far as success in hardening goes, is

in using oil after the thing is cooled in water and immediately at a point where the temperature is reduced so that you can imagine it corresponds with some fixed point on the thermometer, as near as you can imagine it. Be guided by the sizzling of the water, if that will help you, it is an almost sure guide with me.

Now if you will take the steel out at the right moment and drop it into oil and let it cool gradually—let it mellow up, as it were; let the temperature in the centre of the stock distribute itself over the whole mass, you even up the molecular strains and get it into a condition so that you can leave it on your bench for years, and you will not go in some morning and find it accidentally exploded. I think there is much in that point, and I think it is of great practical value in the every-day application of this subject to our needs.

Mr. Porter.—I should add, perhaps, that these springs were tempered in oil, and I would like to say also that a mercury column was used in the shop merely for the information of engineers. English engineers were accustomed to the mercury column and relied upon that altogether, and so for their satisfaction a mercury column was kept in the shop. The Elliotts were makers of barometers on a large scale, and undoubtedly saw to the purity of their mercury. The mercury column in use always corresponded in its indications with the weight. So it must have been pure.

Mr. Rockwood.—Professor Carpenter's paper states that which the experience of almost all engineers proves, that indicators sent out by standard makers are frequently unreliable in their indications.

I lately tested the springs of a pair of indicators in this way. We had a pair of accurate thermometers—obtained in Paris from a celebrated maker—which agreed with each other to within a tenth of a degree. Putting the thermometer into a well in a steam pipe, we read the temperature shown, and finding the corresponding pressure from Porter's Steam Tables, compared it with the pressure as shown by the indicator. I suppose this is a fairly reliable method of testing indicators under steam pressure.

Mr. Thos. R. Almond.—The molecular structure of steel, after this material has been subjected to the disturbing action of rolls, hammer, draw-plate, hardening by fire, etc., is a matter for serious consideration.

Any of these processes disturb the attraction of cohesion. In coiling a wire spring upon a mandrel the material, at the largest diameter of the coil, is stretched, and that at the inside diameter is compressed. If the spring, after being newly coiled, be placed in a bath of acid or alkaline solution, it is probable that the force of capillary attraction is sufficient to allow the liquid to get into the steel and cause fracture, as, by subjecting steel springs to such treatment they will—if under any tension—almost always break. This leads me to believe that many of the unaccountable breakages which occur in machinery are due to similar conditions. In other words, the material may be under sufficient tension to permit the admission of a liquid to an extent sufficient to lessen the attraction of cohesion.

Mr. Barnaby.—I would state that the set which has been described as resulting from dropping a weight on the springs has no relation to the effect which I have explained. The effect which I referred to was the weakening of the springs, and had no reference to a change in length. I always gave the springs a permanent set also, not by dropping a weight on them, but by squeezing them in a vise, giving them a good squeeze so as to compress the coils tightly against each other. The heavy springs always showed a permanent set or shortening as a result of this operation, and as the heavy indicator springs are principally subjected to compression this set may be relied upon as being permanent, as the less than fifteen pounds extension force which they may be subjected to with a vacuum is such a small fraction of the elastic limit that there is no possibility of the set being reversed by it. The light springs being coiled the same diameter and pitch from smaller wire, did not show any set, as the coils would come up solid long before the elastic limit was reached.

After being given this set, the springs very rarely showed any change of length in going through the "jigging" operation. Each spring was carefully calipered for length immediately before and immediately after "jigging." and if any change of length was apparent the spring was thrown out as being improperly tempered. From this it will be understood that there was no change of length as a result of the "jigging" operation, there was merely a change of strength.

In regard to the accuracy of the springs sent out with indicators, I had occasion while working on indicators to test quite

a number of springs at different times, made by other parties, and very rarely found one that was correct. Most of these springs had been used, and usually showed weak under test, which indicated that they had been weakened by use, but as an occasional spring would run two or three pounds strong in forty, while some ran much weaker than could be accounted for by any results I had obtained by the "jigging" process, it is evident that much of the inaccuracy was due to careless adjustment.

Mr. Porter.—I suppose I should defend the method of finishing the indicator springs by polishing the surface with an emery cloth. We found it impossible to obtain exact accuracy by turning the loose head back and forth. The plan finally adopted was a necessity; that was to set the head so as to make the spring a trifle strong, and then reduce the strength by polishing the surface uniformly. We polished every coil equally. It did not affect one coil more than another. It was necessary, in order to bring the spring exactly to the proper strength.

As to the matter of permanent set, that of course was eliminated from the spring in the process of manufacture. I should say, in respect to the springs to which Mr. Barnaby referred, that in this country, before the Thomson indicator began to be made, the indicators made by the American Steam Gauge Company of Boston were simply worthless. They had no pretence to accuracy whatever. The first accurate instruments made in this country were those to which Mr. Barnaby has referred.

*Prof. R. C. Carpenter.**—My attention was called by Mr. Marks, one of the authors of the paper, to the fact that in the calibration diagrams, shown in Fig. 114 and Fig. 115, negative friction is indicated in two of the four diagrams. It would seem that an explanation is necessary to account for this seeming discrepancy. The facts are these: those diagrams were merely shown, in connection with the curves in Fig. 116, as illustrations of the method of representing the spring errors. In this case the log of the test, not published in the paper, shows that the pressures up and down were not exactly the same, and that the difference in steam pressures causes some slight irregularity in the lines, which accounts for what might be considered an abnormal condition, in one or two places.

The discussion has brought out many valuable facts with

*Author's Closure, under the Rules.

reference to the behavior of springs, to which I can add little or nothing from my own personal experience. It is, however, perhaps well known that experiments have been made at the Watertown Arsenal regarding the effect of temperature on the strength of various materials. The principal results of these tests will be found in the published report of the Arsenal for 1888. In this report some few things are collected together, so that the general results are readily perceived, but, on the whole, the information is presented in the shape of individual logs, which can be utilized only after considerable labor and calculation.

With help of Mr. Geo. B. Preston, I have collected the results of the series of tests made on several samples of steel, and computed the corresponding moduli of elasticity. As these results are of considerable interest, and have a direct bearing on the subject of this paper, I will give the more important experiments, so far as they relate to the moduli of elasticity.

In the following tables are given a load per square inch, which is well within the elastic limit, the corresponding total deflection for a length of five inches, the temperature, and the resulting moduli of elasticity :

MILD STEEL.

ANALYSIS.—CARBON 0.09. MG. 0.11.

Load per square inch.	Temperature of specimen, Deg. F.	Total Elongation.	Elongation per inch.	Modulus of Elasticity, Millions.
30,000	0	.0045	.0009	33.3
30,000	70	.0047	.00094	31.8
30,000	201	.0048	.00096	31.3
30,000	306	.0032	.00064	47.0
30,000	384	.0022	.00044	68.0
30,000	437	.0024	.00048	62.5
20,000	545	.0010	.00020	10.0
20,000	668	.0032	.00064	2.1
10,000	736	.0019	.00038	2.6
CARBON 0.2. MG. 0.45.				
30,000	0			
30,000	0	.0073	.00146	22.7
30,000	70	.0044	.00088	34.0
30,000	195	.0026	.00052	57.5
20,000	334	.0011	.00022	91.0
20,000	460	.0046	.00092	21.8

MILD STEEL.—*Continued.*

CARBON 0.31. MG. .57.

Load per square inch.	Temperature of specimen, Deg. F.	Total Elongation.	Elongation per inch.	Modulus of Elasticity. Millions.
30,450	0	.0045	.0009	34.0
30,000	70	.0049	.00098	30.6
30,000	223	.0047	.00094	32.0
30,000	339	.0048	.00096	31.4
30,000	569	.0028	.00092	32.6
20,000	651	.0066	.00092	21.8
20,000	738	.0042	.00084	23.8
20,000	832	.0039	.00074	27.0
10,000	960	.0017	.00034	29.0
10,000	1,021	.0028	.00056	17.7

CARBON .57. MG. .93. LI. .07.

50,000	5	.0092	.00184	27.3
50,000	70	.0082	.00164	30.6
50,000	214	.0085	.0017	29.5
50,000	234	.0073	.00146	34.2
50,000	315	.0068	.00136	36.8
50,000	342	.0060	.0012	41.6
50,000	373	.0054	.00108	46.4
50,000	422	.0086	.00172	29.2
40,000	540	.0083	.00166	24.0
40,500	587	.0075	.0015	26.5
40,000	676	.0085	.0017	23.5
40,000	801	.0070	.0014	28.5
40,000	845	.0081	.00162	24.5
40,000	953	.0118	.00236	16.8

CARBON .97. MG. .8. LI. .28.

50,000	70	.0080	.0016	31.3	30.3
50,000	70	.0085	.0017	29.4	
50,000	265	.0088	.00176	28.4	
50,000	384	.0133	.00266	18.8	
50,000	405	.0118	.00236	21.2	
50,000	478	.0089	.00178	28.1	
60,000	478			25.0	
50,000	505	.0167	.00334	15.0	
50,900	535	.0101	.00202	24.6	
50,000	656	.0187	.00935	5.3	
50,000	736	.013	.0065	7.7	
50,000	867	.0160	.0080	6.2	
50,000	908	.0168	.0084	5.9	
30,000	1,116	.0153	.0076	3.9	

The experiments published do not relate to any steels high in carbon, although the last series of tests relate to steel which approximates at least to spring steel in its composition.

A study of the tables submitted shows very clearly that the temperature effects depend to a great degree on the composition. On the specimens low in carbon the effect of increase in temperature is at first to raise the modulus of elasticity, which possess a maximum in the region of 600° Fahr.; from this point its value rapidly falls off. As the percentage of carbon increases, this maximum occurs at a lower temperature, and in the specimen approaching spring steel in composition, the maximum value is found at a temperature of 70° Fahr.

The tests made on this steel are shown by the line *A B*, in the

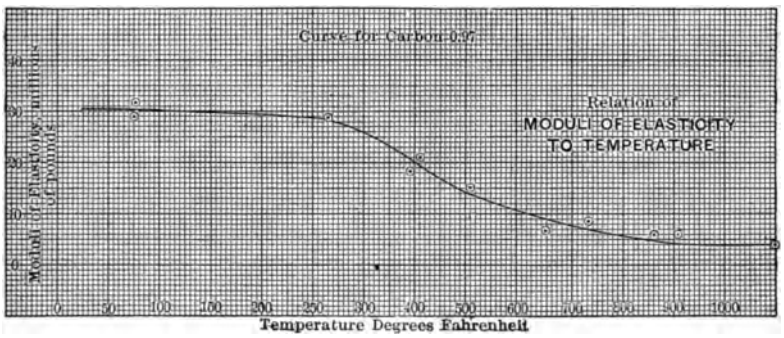


FIG. 127.

accompanying diagram (Fig. 127). Taking the values from the curves we have modulus of elasticity at 60° Fahr. is 30.3 millions, at 206° Fahr. is 29.2 millions. The change in this case is 1.1 millions, or 3.7%, due to a change of temperature of 160° Fahr. (*i. e.*, from 60° to 220°).

While there may be some reasonable doubt as to the exact numerical values, because of irregularity, given in the foregoing tables, there can be no doubt as to the general conclusions, which may be stated as follows:

First.—Change of temperature has great effect in changing elasticity of steel.

Second.—The amount and character of change is dependent upon the composition of the steel. In soft steel, with low carbons, the modulus of elasticity is at first rapidly increased,

passing a maximum at some temperature. and then rapidly diminishing.

Third.—As the carbon increases the modulus of elasticity reaches a maximum at a correspondingly lower temperature, and for ordinary spring steels the maximum value corresponds closely to temperatures of 70° Fahr., so that the effect of heating will tend to weaken any springs made from such steel.

DLXXV.*

TEST OF A BOILER USING GRATES WITH SMALL PERCENTAGE OF OPENINGS.

BY FREDERICK A. SCHEFFLER, NEW YORK CITY.
(Member of the Society.)

OCCASIONALLY there will crop up, in the experiences of engineers, results which seem to be beyond any expectation of what might occur under the circumstances of the investigation being made. The writer has a distinct recollection of a paper having been presented to the Society by the able Past President, Prof. John E. Sweet, entitled "The Unexpected which often Happens," and this paper naturally belongs to the Society's record covering such matters.

The writer had occasion to be called in as a disinterested party some time ago, to make a test of a boiler in use at the American Cereal Mills, Akron, Ohio, the primary object of the test being to ascertain the evaporation per pound of coal which these boilers would produce at their rated horse-power. Upon arrival at the grounds and after making a careful inspection of the arrangement of the boiler, connections, and setting, and of the grates, together with the coal which was to be used during the test, it was found that the grate bars were constructed with such extremely small openings as to lead the writer to almost believe such openings would prohibit them from being used in a satisfactory manner. The kind of coal in use was the ordinary Pittsburg "run of mines," but during the test it was intended to use only the lump part of the coal. The grates in question were of the type shown in Fig. 137, and while the openings call for $\frac{3}{16}$ inch on the drawing, they were not any more than $\frac{1}{4}$ inch, due to rapping the pattern in the mould, and rough edges naturally resulting from the endeavor to make such small openings. These grates had a total area for passage of air of only 15%. The usual size opening for grates burning the kind of coal used is between $\frac{3}{8}$ and

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

½ inch, giving an area of from 35% to 45%. The draught gauge showed .46 inch of water, which, while being a very fair draught, was nothing to brag about, considering the grate openings. The writer felt that it would not be politic to make a test of the boiler under the circumstances, but as everything was prepared at that time to complete the test, it was finally decided to make same, if for no other reason than to demonstrate what the boiler could do with only 15% openings in the grates.

The following results are presented to the Society, as they

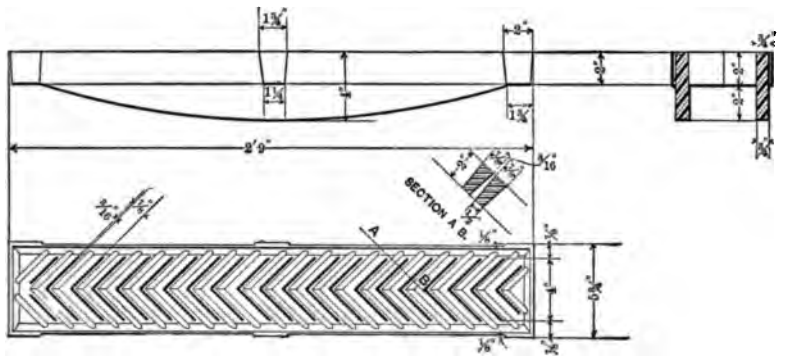


FIG. 137.

seem to be entirely new, and because they appear to be of sufficient interest to be placed on the records.

Figure 138 shows a cross section of the boiler setting, with the grate area, size of ash doors, and draught outlets. In explanation of the results of the test, I would add that there were two boilers in the battery, but as the load was not sufficient to be carried by both boilers, and have each boiler operated near its normal rating, the manufacturers preferred to have the entire load carried by one boiler during the test. Consequently it must be borne in mind, when considering the results, that the boiler was overloaded to a considerable extent; in fact, the actual overload being 60%.

The stack was 6 feet in diameter and 125 feet high. The theoretical draught which can be produced by this stack, with the average air temperature outside of boiler-room 50° Fahr., and temperature of gases 553° Fahr., would be .88 of water. The temperature of the flue was taken close to the boilers, and not in the stack itself, and taking the friction of the gases into con-

sideration, and the fact that there were other boilers connected with the stack, the gases of which were undoubtedly of much lower temperature, the average draught as shown by the gauge approximates closely the theoretical draught.

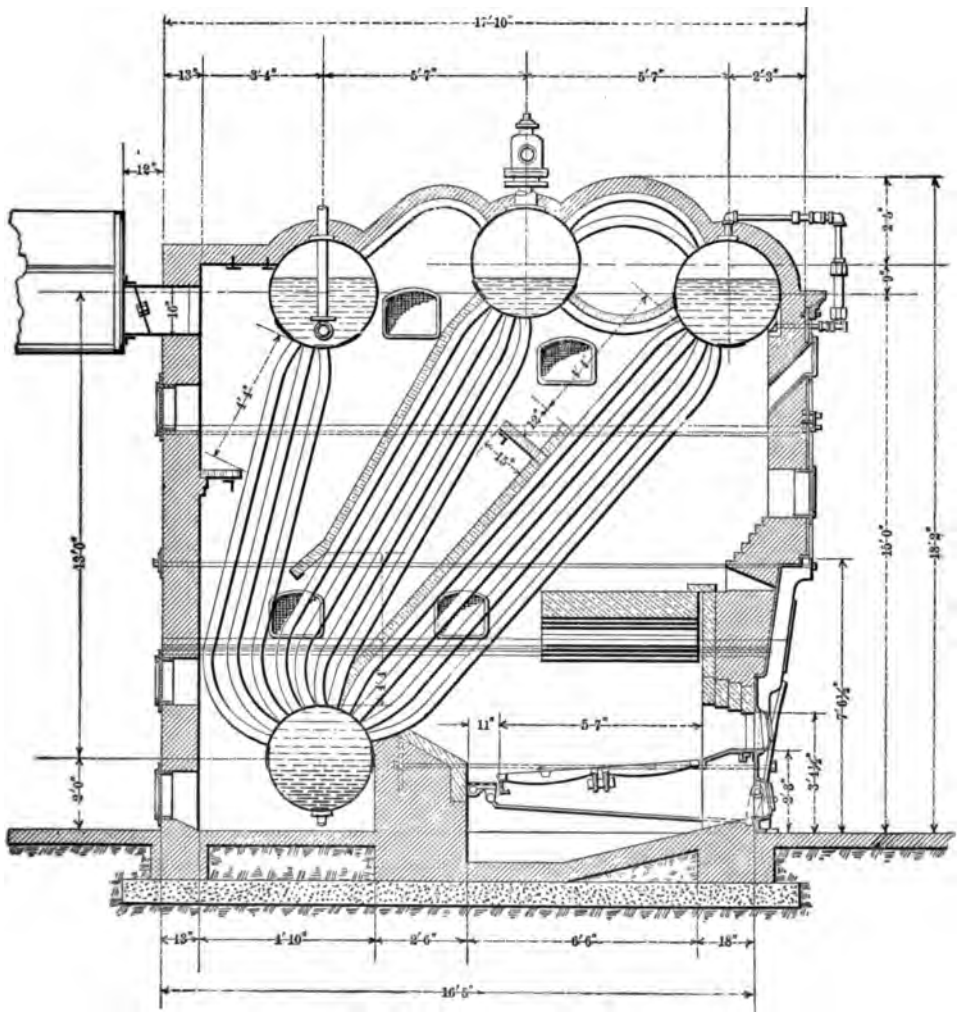


FIG. 138.

In this particular case, no analysis of the coal was made, but a fair average of the theoretical efficiency of the coal used is 13½ lbs. of water from and at 212°.

The results show that, with only 15% grate area, and with an

overload of 60% above the rated capacity, when burning 24.8 lbs. coal per square foot of grate surface per hour, the efficiency obtained was 71%.

Assuming that the volume of air required for theoretical consumption per pound of fuel was 128.4 cubic feet, the total volume of air required per hour would be $128.4 \times 24.8 \times 52 = 165{,}636$ cubic feet at 62° temperature.

I have used the following formula to determine what quantity of air would flow through an orifice similar to the grate openings and having 15% of the total area of the grate surface:

$$V = 352 \sqrt{1 + \frac{t - 32}{493} \times \frac{h}{p}} \quad . \quad . \quad . \quad . \quad . \quad (1)$$

In which V = velocity of air in feet per second.

In which t = temperature of air.

In which h = difference in pressure, or height of column of water in inches.

In which p = barometric pressure.

By substitution, if air is of temperature 62°, the formula becomes:

$$V = 367 \sqrt{\frac{h}{p}} \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (2)$$

and, again substituting values .46 inch for h and 29.35 inches for p, $V = 58.72$ feet per second.

Total grate area being 52 square feet, the area of the openings would be $52 \times .15 = 7.8$ square feet. The quantity of air, therefore, which would pass through this area per second at 58.72 feet per minute would be 458 cubic feet. This amount, however, should be multiplied by a coefficient, which I have called .90, due to the shape of the grate openings. This would reduce the theoretical quantity of air from 458 to 402 cubic feet per second; and per hour the quantity would be 1,447,200 cubic feet. This is about $8\frac{1}{2}$ times the amount actually required for the proper combustion of the coal, which shows that the openings of grates were sufficient to furnish the requisite amount of air, even had the damper been partially closed.

1. Duration of test	6¼ hours.
2. Grate surface, 6 feet 6 inches × 8 feet each	52 sq. ft. each.
3. Heating surface (tube surface only)	2,038 sq. ft. each.
4. Percentage of air opening in grates	15%
5. Kind of coal used	Run of mines
6. Cost of coal per ton (2,000 lbs.)	$1.25

TEMPERATURES AND PRESSURES.

7. Average temperature of flue by pyrometer	553° Fahr.
8. Average temperature of feed-water	38.7° Fahr.
9. Average temperature of air outside of boiler room	50° Fahr.
10. Average pressure of steam by gauge	103½ lbs.
11. Average pressure of draught by gauge	.46 inch

COAL AND WATER.

12. Total coal consumed	8,128 lbs.
13. Total refuse	877 lbs.
14. Net amount combustible	7,251 lbs.
15. Percentage of refuse	10.7%
16. Total amount of water evaporated	56,970 lbs.
17. Equivalent amount of water from and at 212° (factor 1.221)	69,560 lbs.
18. Water evaporated under actual conditions, per pound of coal	7.00 lbs.
19. Water evaporated under actual conditions, per pound of combustible	7.85 lbs.
20. Water evaporated from and at 212°, per pound of coal	8.55 lbs.
21. Water evaporated from and at 212°, per pound of combustible	9.59 lbs.
22. Water evaporated per hour from and at 212°	11,130 lbs.
23. Amount of water evaporated per square foot of heating surface per hour, from and at 212°	5.46 lbs.
24. Amount of water evaporated per square foot of grate surface per hour, from and at 212°	214 lbs.
25. Amount of coal burned per square foot of grate surface per hour	24.8 lbs.
26. Amount of water evaporated per $1 worth of fuel, from and at 212°	13,680 lbs.

HORSE-POWER.

27. Horse-power on builder's rating	200
28. Horse-power on Centennial standard, 34½ lbs. of water from and at 212°	322
29. Per cent. above builder's rating	60%

DISCUSSION.

Mr. William Kent.—To design grate-bars with only 15% of air space would not ordinarily be considered good practice; but after Mr. Scheffler's test has showed that under the conditions of that test they worked well enough, we may ask why should they not work? A common rule for proportioning draught openings is $\frac{1}{6}$ of the grate surface for area over the bridge wall, $\frac{1}{7}$ through the flues, and $\frac{1}{8}$ in the chimney, decreasing as the temperature decreases. As the absolute temperature of the air passing through the grate-bars, and also its volume, is less than half of that of the gases in the flues, why should not air openings of $\frac{1}{15}$ the grate surface, or half that of the flue area, be sufficient, instead of 15%, as in Mr. Scheffler's test; or 30% to 50%, as in ordinary practice? The fact is, that, when the grate-bars are free from clinker, they rarely, if ever, choke the draught to any appreciable extent, the real choke of the draught being in the pile of coal lying on the grate-bars. The amount of this choke varies through an exceedingly wide range from total obstruction, as when a partially coked layer forms on top of a freshly fired portion of soft coal, when too much has been fired at a time; to too great freedom from choke, as when anthracite egg-coal is fired in thin layers, allowing so much air to pass through the fire as to chill the gases of combustion and destroy the economy.

Although Mr. Scheffler's test shows that under some conditions 15% of air opening through the grates is sufficient, it will still continue to be good practice to construct them with 50%. The increased per cent. of opening costs nothing; it may be useful if the coal has a tendency to clinker; and if at any time it should be advisable to choke the passage of air through the opening, it can be done to any extent desirable by carrying thicker fires, or by partly closing the damper in the chimney.

Mr. W. B. Le Van.—Was any analysis made of the gases?

Mr. Scheffler.—No, sir.

Mr. A. A. Cary.—Was any test made of the quality of the steam supplied by a boiler evaporating this quantity of water? Why was there no calorimeter test made?

Mr. F. A. Scheffler.—No calorimeter test was made, because the facts for which I was in search at that time did not demand

such a determination, but I have tested other boilers set exactly like this one, and the calorimeter test showed ½ of 1% of moisture, which I would consider to be as good as any other boiler could obtain. That test was made under circumstances where the boiler was forced worse than it was in this particular case, because we used fuel oil and were very much crowded for the power, and had to obtain about 40% or 50% excess of the rated capacity of the boiler, and fuel oil is exceptionally hard on the first rows of tubes which take the heat from the flames.

DLXXVI.*

SOME EXPERIMENTS ON THE EFFECT OF WATER HAMMER.

BY R. C. CARPENTER, ITHACA, N. Y.,
(Member of the Society),
AND S. HENRY BARRACLOUGH, B.E., SYDNEY, AUSTRALIA.†

AN examination of all the available literature indicates that very little experimental work has been done on the subject of water hammer, and an equally small amount of theoretical consideration has been given it. In Vol. XIV. of the Proceedings of the American Society of Civil Engineers there is an account of some tests made at the water works at Providence, R. I.; but the velocity in the mains where the force of the ram was measured was not large, and, although the tests were interesting in themselves, no attempt was made to deduce any theoretical results therefrom. Again, in an issue of *Mechanics*, August, 1884, Mr. Nystrom discusses some cases of water hammer, making use in his analysis of the fact that the momentum of the water in the pipe at the initial instant of closing the valve is completely exhausted at the end of that operation. He here assumes that the resistance met by the mass of moving water is constant, so that his result gives only the average pressure due to the arrest of momentum, and not the maximum. Lastly, there are some interesting articles by Professor Church in the *Journal of the Franklin Institute* for 1890, in which the writer traces the variation of the pressure just above the valve gate for certain definite cases, with numerical examples. It is there shown that the maximum pressure produced in a pipe by the closing of a valve depends not only on the time taken to close it, but also on the manner

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† The experiments described in the following paper were performed by Mr. S. H. Barraclough, B.E., and A. W. Wigglesworth, M.E., graduate students in Sibley College, at my request, during the winter and spring of 1892–93.—R. C. CARPENTER.

of doing so; or, in other words, the motion of the gate may be managed in such a way that the maximum pressure shall be much less than with some different management, the whole time of closing being the same in the two cases.

METHOD OF TESTING.

The first set of observations was taken on the water hammer which is produced in a pipe $1\frac{1}{2}$ inches in diameter by the sudden closing of an ordinary self-closing basin tap, as shown in Fig. 143, of a size fitting $\frac{1}{2}$ inch pipe. After several trials of different methods for measuring the pressures produced, a Crosby indicator was finally used, with a 40 lb. spring. The only difficulty encountered in the use of the indicator was the method of giving a suitable motion to the drum. It was found impracticable, with the means at hand, to give it an accurate uniform motion, and consequently it was given a uniformly accelerated motion with the aid of the drum spring of the indicator itself. Before making each test the drum was pulled out to its farthest extent, and then it was allowed to run back, as it did so causing a small flywheel to which its cord was attached to rotate; the indicator card was taken during this return motion. The assumption is here made that the drum spring exerts a uniform pull at all portions of its stroke. Even if this assumption is not perfectly accurate, it is sufficiently true for the purpose of the present investigation. The time taken for the full turn of the drum was found by means of a chronograph owned by Sibley College, and, therefore, on the above assumption of a uniform pull, the velocity of the drum at any particular point is determinable. The average length of line drawn on the chronograph drum in one second was found to be 2.15 inches, while the average length drawn during the time taken for the full travel of the indicator drum was 2.63 inches; or, in other words, the time taken for the full travel of the drum when moved by its own spring was $\frac{2.63}{2.15}$ or 1.23 seconds. The distance on the indicator card corresponding to this motion was 5.28 inches, so that the values of s (the space) and t (the time) in the equation

$$s = \tfrac{1}{2} f t^2$$

are now known, and f may therefore be calculated; and hence

for any given value of s the corresponding time will then be known.

Points, through which the accompanying curve (Fig. 146) was drawn, have been obtained in this way, and the curve was used to determine the time of any particular phenomenon shown on the indicator cards.

METHOD OF OBTAINING DIAGRAMS.

The indicator and water tap were attached side by side to a small water-main of $1\frac{1}{2}$ inches internal diameter, the static pressure in which was about 30 lbs. per square inch. The actual diagrams obtained are shown in Figs. 139 to 142. The successive waves produced are clearly illustrated, there being several distinct ones in each case. Several results may be obtained from the diagram, as, for instance, the time taken in closing the tap. The point a (see Fig. 139) marks the extreme limit of movement of the indicator drum, and corresponds to the zero or origin of diagram A. At b the tap begins to close, and is entirely closed at d. By dropping a perpendicular dc from d, and measuring the difference on the diagram of the ordinates corresponding to the abscissæ ab and ac, the time corresponding to bc is obtained, and this is the time taken to close the valve. In the present instance it does not exceed $\frac{1}{100}$ of a second.

Similarly, by dropping perpendiculars from each of the wave crests, the time between each successive pair of waves may be obtained, and it can thus be determined whether the waves are isochronous or not. This was done on several cards, showing that the waves are practically isochronous, though the method used is not accurate enough to settle the matter with absolute certainty.

At the time of taking the diagram (Fig. 139), there was an air chamber on the main, some distance from the indicator and tap. This was afterward removed, and the diagram shown in Fig. 140 taken, in order to see what effect the air chamber had on the water hammer. This chamber was then filled with water, and diagram (Fig. 141) taken. This is a practical case, since ordinary air chambers when in use often become full of water. The cards shown in Figs. 140, 141, and 142 were taken one after the other, and represent the case of no chamber at all, an air chamber, and a "water chamber" respectively. As might be expected, the

smoothest curve, and that showing least maximum pressure, appears when the air chamber is used, while the most irregular card is produced by removing the chamber altogether, the water

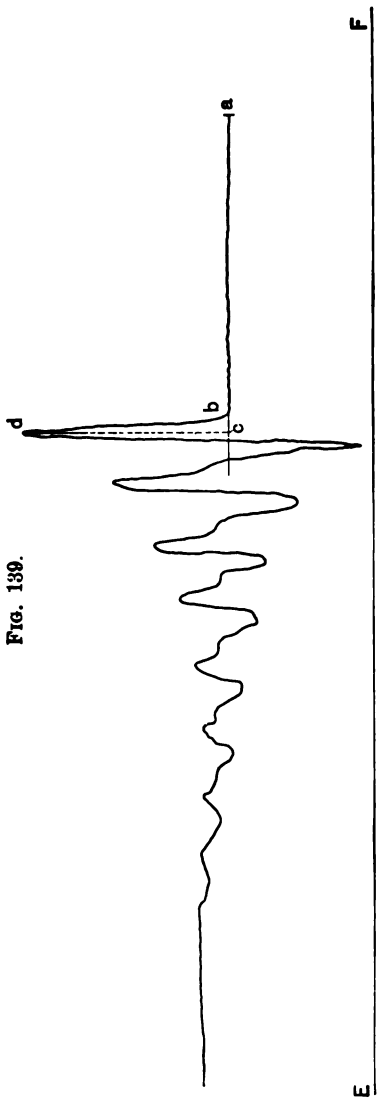


Fig. 139.

DIAGRAM OF WATER HAMMER, 1½ INCH PIPE, NO AIR CHAMBER

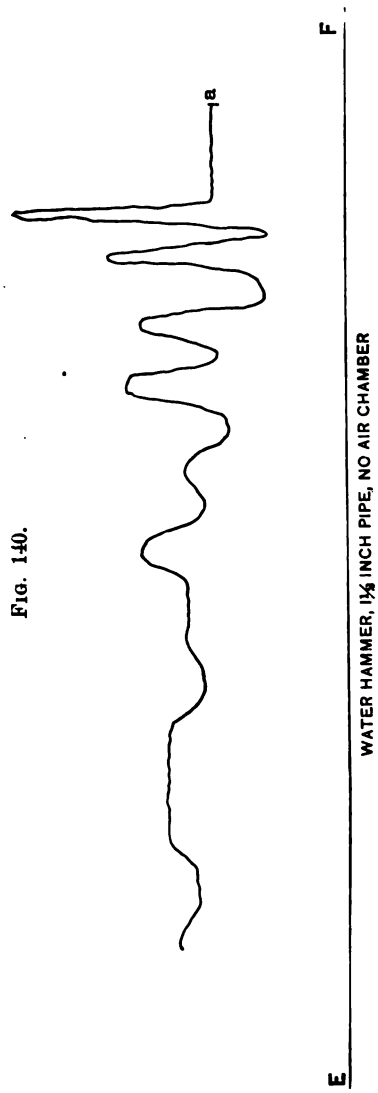


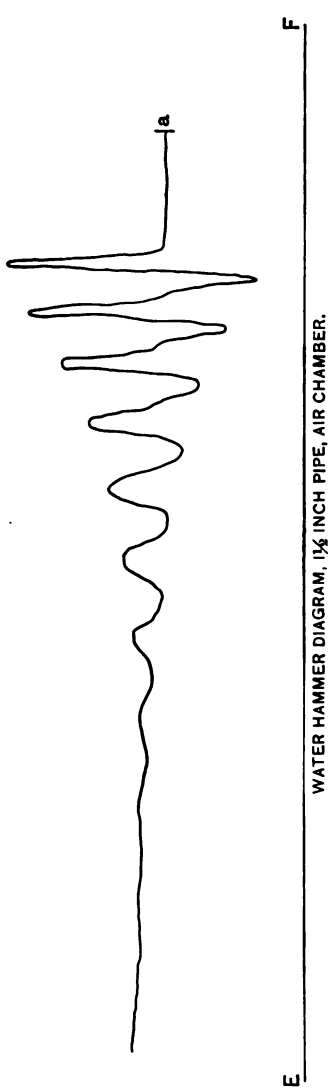
Fig. 140.

WATER HAMMER, 1½ INCH PIPE, NO AIR CHAMBER

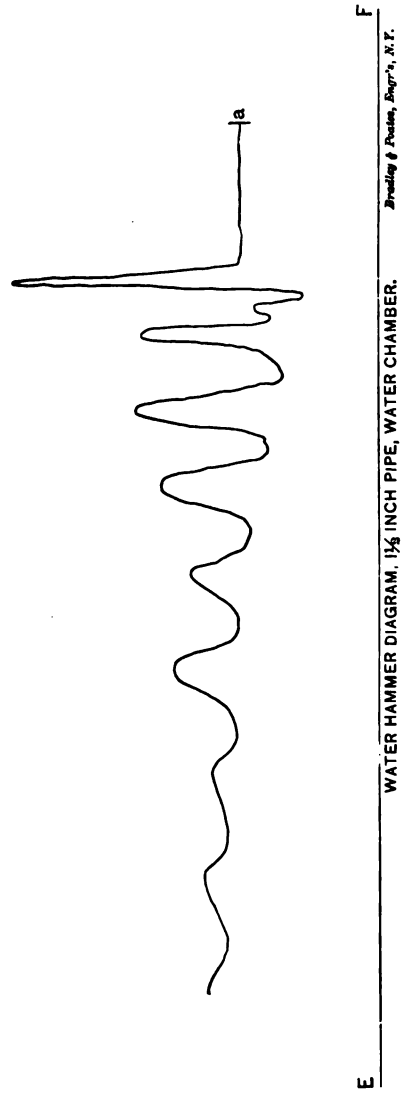
chamber producing an effect intermediate between the other two.

Several other sets of cards were taken at different times and

under different velocities of flow, the differences from the set shown in each case being small, and due, probably, to variation in time of closing the tap.



WATER HAMMER DIAGRAM, 1½ INCH PIPE, AIR CHAMBER.
FIG. 141.



WATER HAMMER DIAGRAM, 1½ INCH PIPE, WATER CHAMBER.
FIG. 142.

The average pressures are essentially as given in the following table. The pressure is expressed in pounds above atmosphere.

TABLE SHOWING EFFECT OF WATER HAMMER IN 1¼-INCH PIPE, PRODUCED BY OPENING AND CLOSING A ¼-INCH TAP.

	NO AIR CHAMBER.		Air Chamber.	Air Chamber filled with water.
	Fig. 1.	Fig. 2.		
Static pressure	29.5	28.5	27.5	28
No. of distinct blows	8	9	9	9
Maximum pressure	72.5	69.0	61.5	76.0
Minimum pressure	2.5	16	10	9
Time pulsations continue	0.8 sec.	1.2	0.8	1.1 sec.
Pressure at end of one second	36	36	31.5	36
Ratio of increase of pressure	2.47	2.56	2.15	2.70

A study of the diagrams shows that the wave phases are more regular in case of the air chamber, and have a regular form distinct from and different from the cases where no air chamber

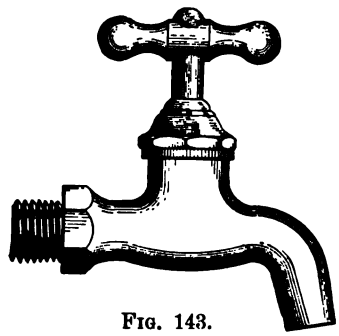


FIG. 143.

was used. In the case of a water chamber, or of no air chamber, the wave motion is decidedly irregular, and the form quite different from that in the preceding case.

WATER HAMMER PRODUCED BY FULL-SIZED OPENING.

These tests, although interesting, were not considered satisfactory, owing to the low velocity of water in the main, the area of which was large compared to the size of the tap orifice, and a new set were undertaken in which the hammer was produced by suddenly closing a gate valve across the full section of a 2-inch water main. By this means the test was made with a much greater velocity than in the previous case, and the conditions

corresponding more nearly to the pressures produced in actual practice.

The following sketch (Fig. 144) illustrates the arrangement of the apparatus used:

A is the valve through which the water enters, and by which it can be throttled so as to vary the pressure from 30 lbs. to 5 lbs. per square inch.

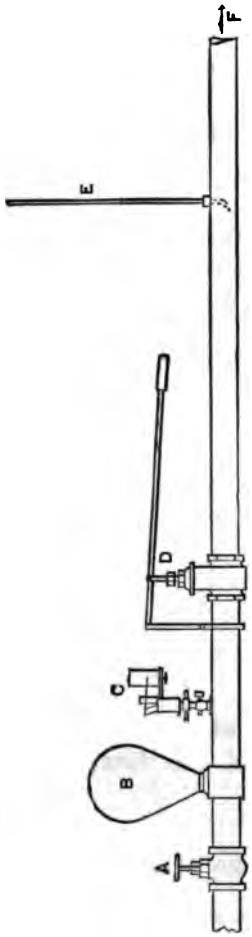


Fig. 144.

B is an air chamber. When it was desired to make a test without any chamber, it was unscrewed, and a plug put in its place. The "*water chamber*" was constructed by turning *B* on the pipe so that it pointed vertically downward.

C is a Tabor indicator for measuring the pressures produced. A 150 lb. spring was used. The method of moving the drum, and the characteristics of the card are explained below.

D is the throttle valve used for suddenly arresting the progress of the water. It was made by cutting the thread off the spindle of an ordinary gate valve, so that the valve could be shut almost instantaneously by means of the lever attached to the top of the spindle. *E* is a Pitot tube for measuring the velocity of the water in the main. This was calibrated before making the tests, by observing the heights which the water rose in the tube, and noting the corresponding times which it took for a measured volume of water to flow through the mouthpiece *F*. The attached curve (Fig. 147) shows the results of this calibration. The Pitot tube was used to determine the velocity, instead of observing the amount of water which passed at each operation, on account of the great saving in time thus effected; and although this method is not quite as accurate as the other, yet it is evident, from the regular character of the curve, that the results obtained by its use will be a fair approximation to the truth.

With regard to the indicator diagrams obtained, it was decided that although there were many interesting points in the subject to be determined, yet the one of most practical importance was

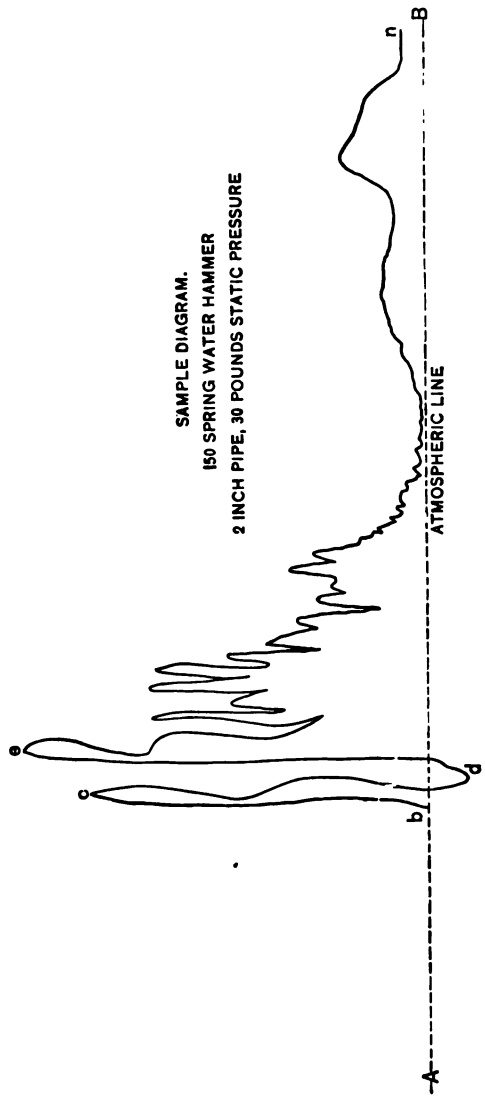


FIG. 145.

the maximum pressure produced under any particular set of circumstances, and so in this series of tests no attempt was made to determine the time value of the horizontal dimensions of the

card, and the drum of the indicator was in consequence only moved by hand.

Tests were made with velocities varying from 3 to 9 feet per second, and under the following conditions:

(1) With no air chamber on the pipe at all.
(2) " an " " capacity, 320 cubic inches.
(3) " no " " " 40 " "
(4) " a water chamber " 320 " "

The method of making a test was as follows:

The valve *D* was raised to its full height, and the water allowed to flow for a short time, till its motion became steady. The reading of the Pitot tube was then observed, and the valve *D* suddenly closed, the pressure produced being registered by the indicator.

The water main was stayed underneath, at each end, during the tests, to keep it steady; but the shock produced by closing the valve was so great as to make the whole apparatus vibrate badly, and consequently the diagrams are not nearly so smooth and regular as in the first series of tests; but this was a matter of small importance, as the maximum ordinate was not affected, and that was the only measurement made from the card. An example of the general type of diagram obtained is given in Fig. 145.

It will be noticed that the point of minimum pressure falls some 15 lbs. below the zero line, which can only be caused by the momentum of the moving parts of the indicator. No doubt the same thing occurs at the point of maximum pressure, so that these readings are probably too high by that amount. Possibly the point of minimum pressure would not fall even as low as the zero line were it not for the momentum of these moving parts; but, as in many cases where the maximum pressure is small and the movement of the parts correspondingly slight, the minimum pressure falls to zero, or nearly so, it is probable that only the pressure below the zero line is due to the momentum of the parts.

It is also noticeable that, in nearly every instance, the wave of maximum pressure is the second, and not the first, as would naturally be expected.

The table which follows gives the results of the test, and the accompanying curves (Fig. 148) show the variation of maxi-

mum pressure, with velocity of flow for the cases of no air chamber, and for two different sizes of air chambers. The curve

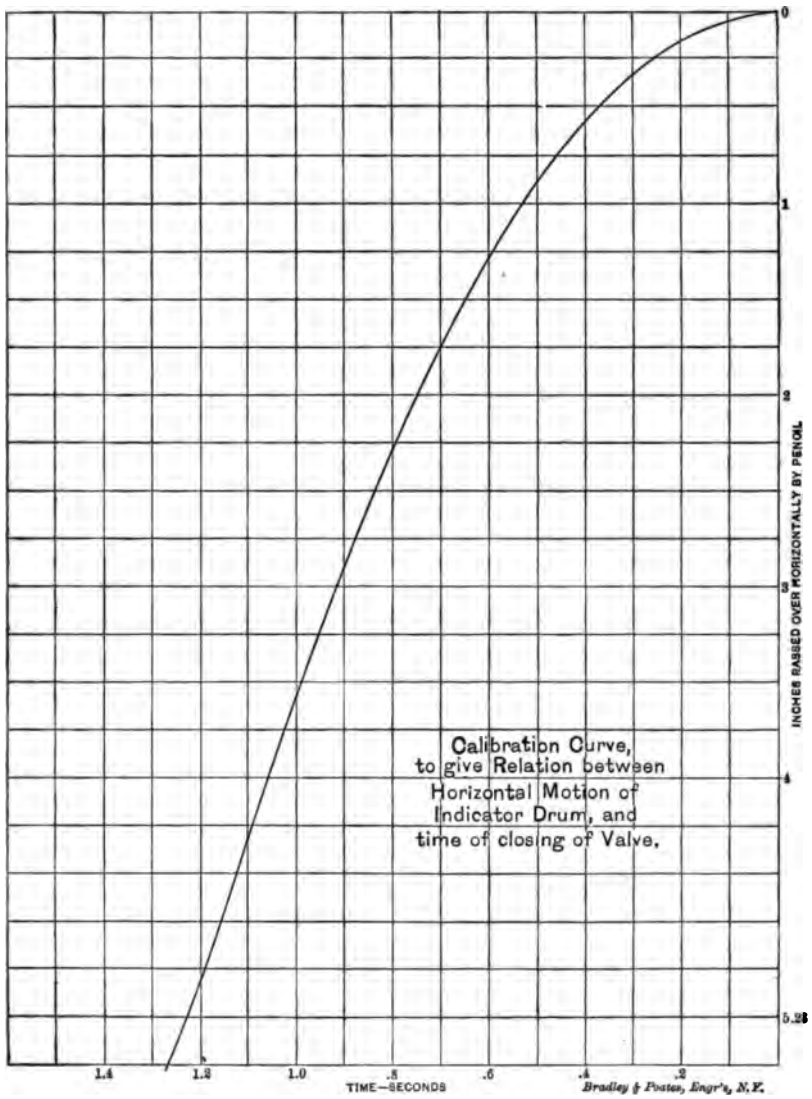


FIG. 146.

for the case of the water chamber is not shown, as it lies close to, and a little above, that for the case of no air chamber at all, and it would only have confused the diagram to insert it.

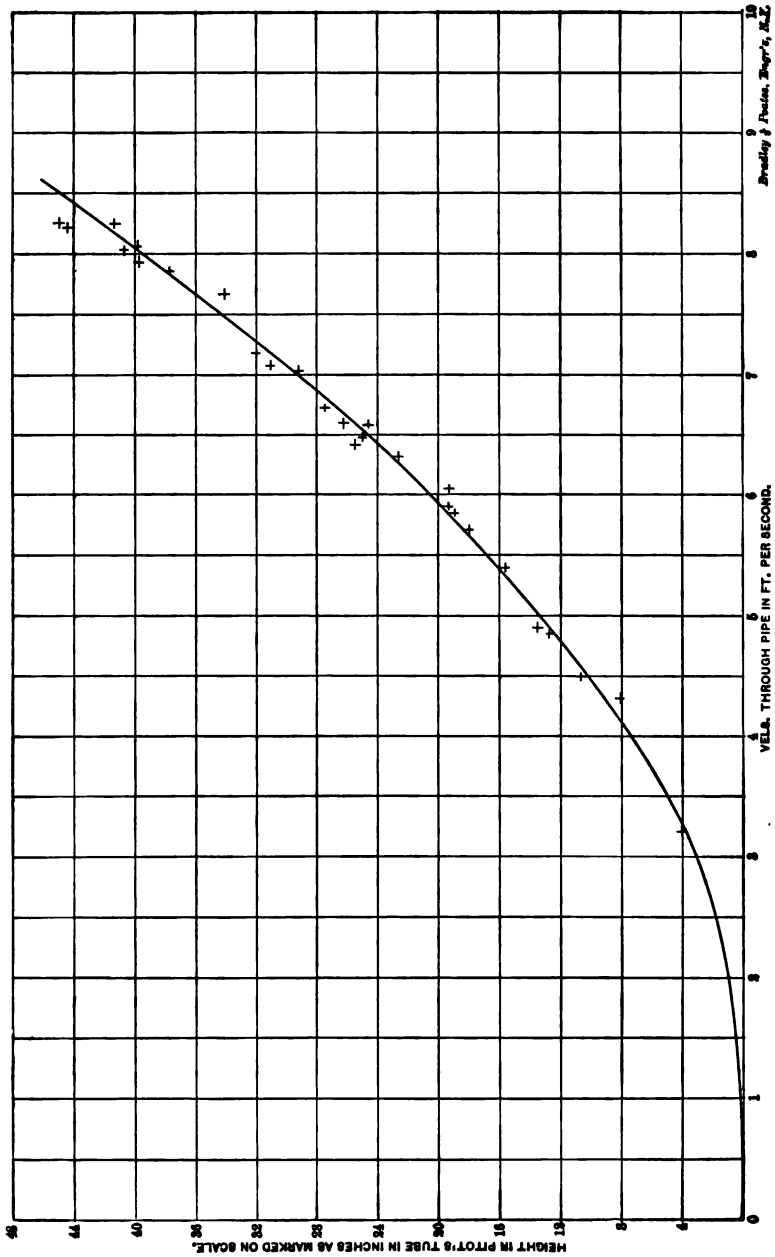

VELS. THROUGH PIPE IN FT. PER SECOND.
HEIGHT IN PITOT'S TUBE IN INCHES AS MARKED ON SCALE.
0
1
2
3
4
5
6
7
8
9
10
4
8
12
16
20
24
28
32
36
40
44
48
Bradley & Foster, Engr's, N.Y.

FIG. 147.

The two remaining curves are fairly uniform in character. All three curves have their origin in the point of static pressure of the water in the main, as this pressure is always acting. The results are interesting, as showing the high pressure that

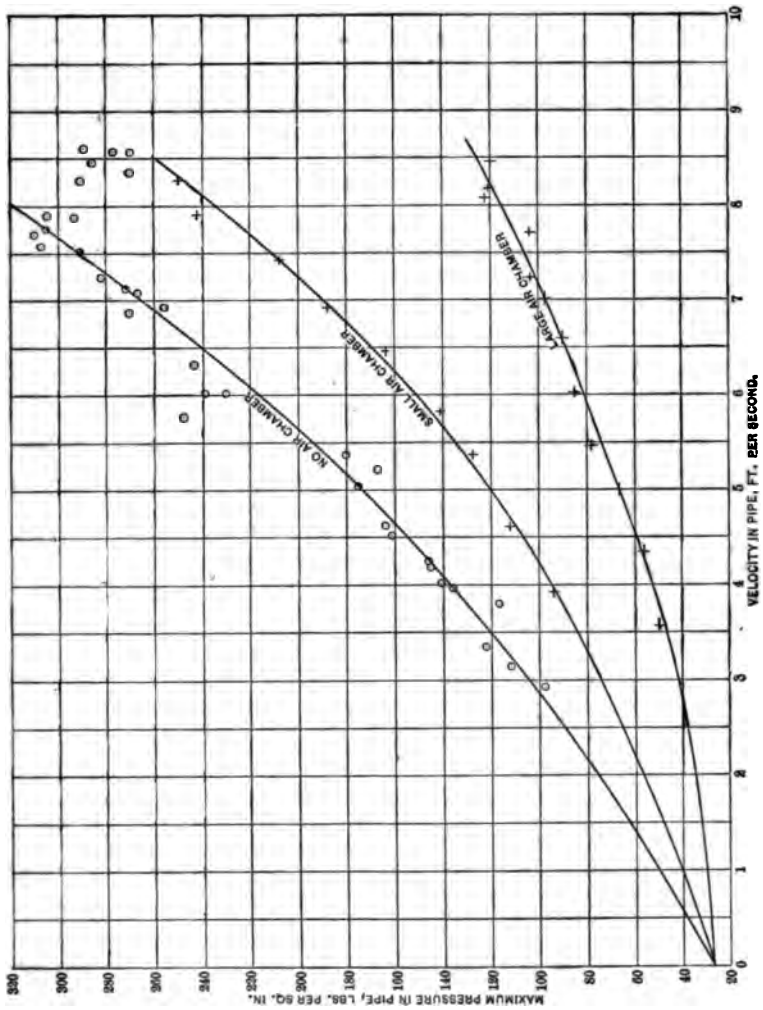


FIG. 148.

may be produced by stopping the flow of water through a pipe, even when the velocity is small. It is also interesting to note the relative effect of the air chambers: although the larger one was eight times the volume of the smaller, it only halved the pressures produced by the hammer.

The present set of experiments do not pretend to have demonstrated any new principle, but, as was stated at the beginning, the amount of experimental work done on the subject is very meagre, and these results may be of assistance to anyone intending to investigate the subject more thoroughly; while they also may be of service to the hydraulic engineer, as showing the maximum possible pressure which can be produced by water ram, under circumstances similar to those surrounding this series of tests.

LOG AND RESULTS OF TEST.

Water Chamber.		No Air Chamber.		Small Air Chamber.		Large Air Chamber.	
Velocity.	Pressure.	Velocity.	Pressure.	Velocity.	Pressure.	Velocity.	Pressure.
3.92	152	2.92	98	3.92	94	3.70	50
5.27	203	3.15	111.5	4.62	112	4.42	56
6.00	230	3.35	122	5.37	128	5.02	66
6.65	242	3.80	117	5.82	141	5.45	78
6.65	256	3.97	136	6.46	164	6.02	85
7.17	271	4.02	141	6.92	188	6.62	90
7.37	297	4.20	145	7.42	208	7.02	97
7.47	294	4.25	145	7 90	242	7.25	103
7.85	295	4.52	161	8.26	249	7.73	104
8.25	304	4.62	164			8.07	122
8.52	315	5.05	175			8.18	120
		5.22	167			8.45	120
		5.37	180				
		5.77	248				
		6.02	230				
		6.02	239				
		6.32	244				
		6.87	271				
		6.93	256				
		7.07	266				
		7.13	272				
		7.25	281				
		7.52	291				
		7.57	307				
		7.70	309				
		7.75	305				
		7.90	305				
		7.87	293				
		8.27	291				
		8.35	270				
		8.55	270				
		8.55	277				
		8.45	286				
		8.60	289				

ADDED AT THE MEETING.

In the experiments described, the mass of water in motion was indefinite, as the pipe on which the experiments were made was connected directly to the university water-works. The total amount of water in motion was, as nearly as could be determined, as follows: 30 feet, 2-inch pipe; 33 feet, $2\frac{1}{2}$-inch pipe; 150 feet, 3-inch pipe; 375 feet of 6-inch pipe. These pipes were supplied by a reservoir about $\frac{1}{4}$ acre in area, and the static head was practically constant at 29 lbs.; the maximum variation not exceeding more than $1\frac{1}{2}$ lbs. in either direction. These are about the conditions occurring in ordinary city water-works.

In the second series of experiments, which are described in the paper, the hammer was produced by closing the valve *D*, shown in Fig. 144; the velocity of the outflowing water was regulated by opening the globe valve *A* a greater or less amount.

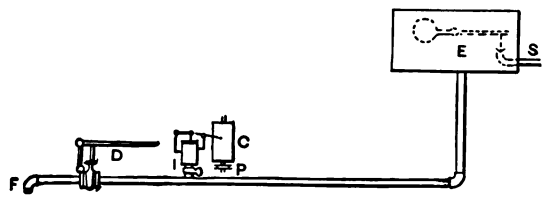


FIG. 149.

With the pressure of 29 lbs. per square inch this method was very satisfactory and gave uniform results, although it may have obscured the effect by addition of unusual conditions.

It was desirable to perform a similar series of experiments under such conditions as would give an accurate knowledge of the mass of water in the system, which evidently contributes to produce the shock. These experiments were conducted by Messrs. Houghton and Spencer, graduate students in Sibley College, under my general supervision, in the following manner:

A 2-inch pipe was led, as shown in the sketch (Fig. 149), to a tank about $2\frac{1}{2}$ feet by 3 feet in area, and 2 feet in depth, which was supplied with water from the main supply pipe by a closet

float-valve. This apparatus provided a head which was quite constant, its maximum variation proving to be less than $\frac{1}{8}$ of a foot. The head being only about 4 feet, it was found necessary to adopt a different method of regulating and measuring the velocity, as the water would not rise of sufficient height in a Pitot's tube to be seen; the plan was adopted of leaving the pipe fully open and regulating the flow by screwing bushings of various sizes into the discharge opening *F*. The discharge was received into a tank and weighed.

The indicator was arranged as shown in the sketch, except that it projected horizontally from the pipe instead of vertically. The diagram was drawn, not on the regular indicator drum, but on an independent drum *C*, which could be revolved at a perfectly definite and regular rate of speed by a cord passing over the pulley *P*, and driven by a motor. This method, although giving low velocities, was no doubt an improvement in every respect over the other, as in calibrating these discharge orifices the different measurements of the time required to pass a given weight of water differed from each other by less than $\frac{1}{8}$ second.

The following are the data and results of the tests made by Houghton and Spencer: Length of pipe, from supply tank to discharge valve, 53 feet 5 inches, $1\frac{1}{2}$-inch pipe, and 4 feet 6 inches of 2-inch pipe. No air chamber. Area of $1\frac{1}{2}$-inch pipe, 2.038 square inches; area of 2-inch pipe, 3.356 square inches = 0.0233 square foot. Circumference of drum on which diagrams were taken, 19.2423 inches. Distance moved on drum in 1 second of time, 1.282 inches. Distance from indicator to discharge valve, 3 feet 6 inches. Weight of water in pipe, 53.75 lbs. for test *A*.

Test A, Set No. 1.—Discharge under 4-foot heat through $\frac{3}{8}$ to $\frac{1}{4}$ inch bushing. Rate of flow, 1 cubic foot in $56\frac{1}{2}$ seconds. Velocity in feet per second in 2-inch pipe, 0.76 feet per second; in $1\frac{1}{2}$-inch pipe, 1.25 feet per second. Diagrams taken with 40 spring.

No. of Test.	Pressures from Atmosphere.			Time in Seconds.	
	Maximum above.	Minimum below.	Second wave above.	Of closing valve.*	Of one vibration.
1	58	10	24	0.023	0.17
2	56	10	25	0.024	0.17
3	60	10	25	0.020	0.17
4	58	9	22	0.023	0.18
5	58	10	23	0.023	0.18
Average.....	58	9.8	22.8	0.0226	0.174

Test A, Set No. 2.—Discharge under 4-foot head, through $\frac{1}{2}$ to $\frac{3}{8}$ inch bushing. Rate of flow, 1 cubic foot in 39 seconds. Velocity in 2-inch pipe, 1,105 feet per second; in $1\frac{1}{2}$-inch pipe, 1.82 feet per second. Diagram taken with 80 spring.

No. of Test.	Pressure from Atmosphere.			Time in Seconds.	
	Maximum above.	Minimum below.	Second wave above.	Of closing valve.	Of one vibration.
1	80	10	32	0.023	0.17
2	92	10	46	.....	0.17
3	88	10	40	.....	0.17
4	90	10	46	.....	0.17
5	88	8	38	.....	0.17
6	90	8	44	0.023	0.17
7	74	8	32	.....	0.17
Average..	86	9.1			

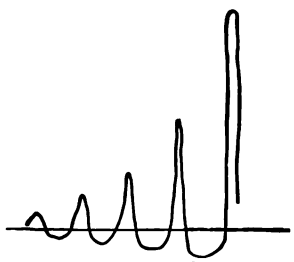


Fig. 150.

Test A, Set No. 3.—Discharge under 4-foot head through bushing, $\frac{3}{4}$ to $\frac{1}{2}$ inch pipe size. Flow, 1 cubic foot in $32\frac{1}{3}$ seconds. Velocity in 2-inch pipe, 1.33 feet; in $1\frac{1}{2}$-inch pipe, 2.2 feet per second.

1	83	10	36	....	0.16
2	92	10	36	....	0.24
3	84	10	30	....	0.16
4	102	10	48	....	0.24
5	88	10	38	0.03	0.18
6	108	12	50	0.02	0.18
7	104	10	39	....	0.18
Average	94.6				

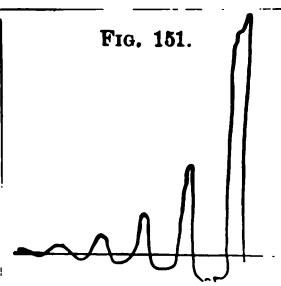


Fig. 151.

* Time of closing the valve in every case was estimated as the difference in the length of the first and succeeding waves. This is to be considered only an approximation of the actual time taken.

Test A, Set No. 4.—Discharge under 4-foot head through bushing, 1-inch to ¾-inch pipe size. Flow, 1 cubic foot in 27.8 seconds. Velocity in 2-inch pipe, 1.54 feet per second; in 1½-inch pipe, 2.55 feet per second.

No. of Test.	Pressure from Atmosphere.			Time in Seconds.	
	Maximum above.	Minimum below.	Second wave above.	Of closing valves.	Of one vibration.
1	103	9	....	....	....
2	114	12	....	0.17	....
3	120	8	38	0.20	0.02
4	120	8	42	0.20	0.02
5	124	....	....	....	....
6	136	....	....	....	....
7	116	....	39	....	....
8	119	12	42	0.17	0.01
9	114	9	40	....	....
10	114	10	42	....	0.03
11	110	10	40	....	0.03
12	100	....	....	....	....
13	114	....	....	....	....
14	100	....	....	....	....
15	106	10	30	0.17	0.03
Average	...114				

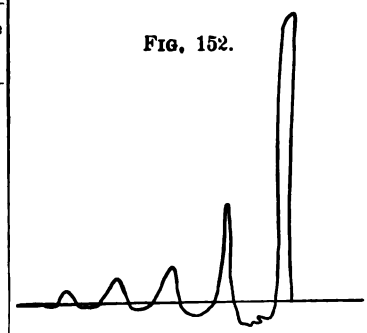


Fig. 152.

Test A, Set No. 5.—Discharge under 4-foot head through bushing, 1¼-inch to 1 inch pipe size. Flow, 1 cubic foot in 25.2 seconds. Velocity in 2-inch pipe, 1.705 feet; in 1½-inch pipe, 2.85 feet.

No. of Test.	Pressure from Atmosphere.		
	Maximum above first wave.	Minimum below.	Second wave above.
1	122	1	30
2	110	1	20
3	104	2	22
4	126	2	26
5	116	2	20
6	106	2	12
7	102	2	10
8	104	2	26
Average.	114	1.75	21

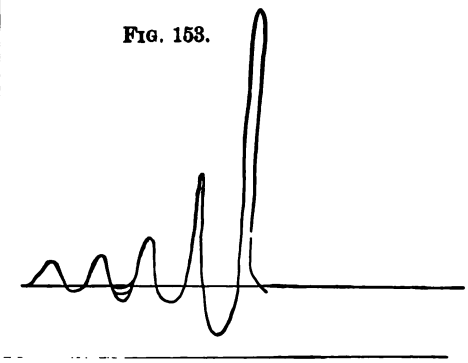


Fig. 153.

In this test, waves not of sufficient number to estimate time interval.

Test B.—Discharge valve moved closer to supply tank, making in the system 20 feet 1½-inch pipe, 4½ feet 2-inch pipe. Arranged as before, except that drum on which diagrams were taken was moved by hand. Weight of water held by the pipes and constituting the mass in motion, 24.3 pounds. The results give the maximum pressures only.

Discharge Bushing.	Velocity, Feet, per Second.		Pressure from Atmosphere.			Time of discharge, one cubic foot.	Test No.
	1½-inch pipe.	2-inch pipe.	Maximum above.	Second wave.	Minimum below.		
						Seconds.	
⅝ to ½	1.29	0.805	22.5	11	5	54.2	*B*—6
⅝ to ½	1.29	0.805	21	7	5	54.2	*B*—6
½ to ⅝	2.01	1.25	61	30	10	34.87	*B*—7
⅝ to ½	2.54	1.58	85	45	8	27.8	*B*—8
1 to ⅝	3.32	2.06	125	55	10	21.1	*B*—9

General Results.

The general results of the two tests are coördinated with respect to maximum pressures and velocities in Fig. 154. The results show in each case an increase in pressure proportional to the velocity. The pressures for test B, in which case 24.3 lbs. of water was in motion, are much less than for test A, in which case 53.75 lbs. of water was in the pipe. This is true for corresponding velocities, but it is to be noted that the pressures for the same velocities are not proportional to the weights. A conclusion drawn from a test involving so few cases cannot of course be general in its nature, as it may be due to special conditions which existed at the time of the test, but supposing the case shown by the diagram a general one, or at least note the deduction from the relative inclination of the lines shown on the diagram. It will be seen that at a velocity of 5.4 feet per second, the pressure in each case would be 270 lbs., or in other words the pressure at the point would be the same in either case. I am of the opinion that this is due to the peculiar alignment of our pipe system, there having been four more elbows in use for test A than for test B.

The equation of the line nearly representing the tests made in case A would be

$$p = 41.5\,v + 8.$$

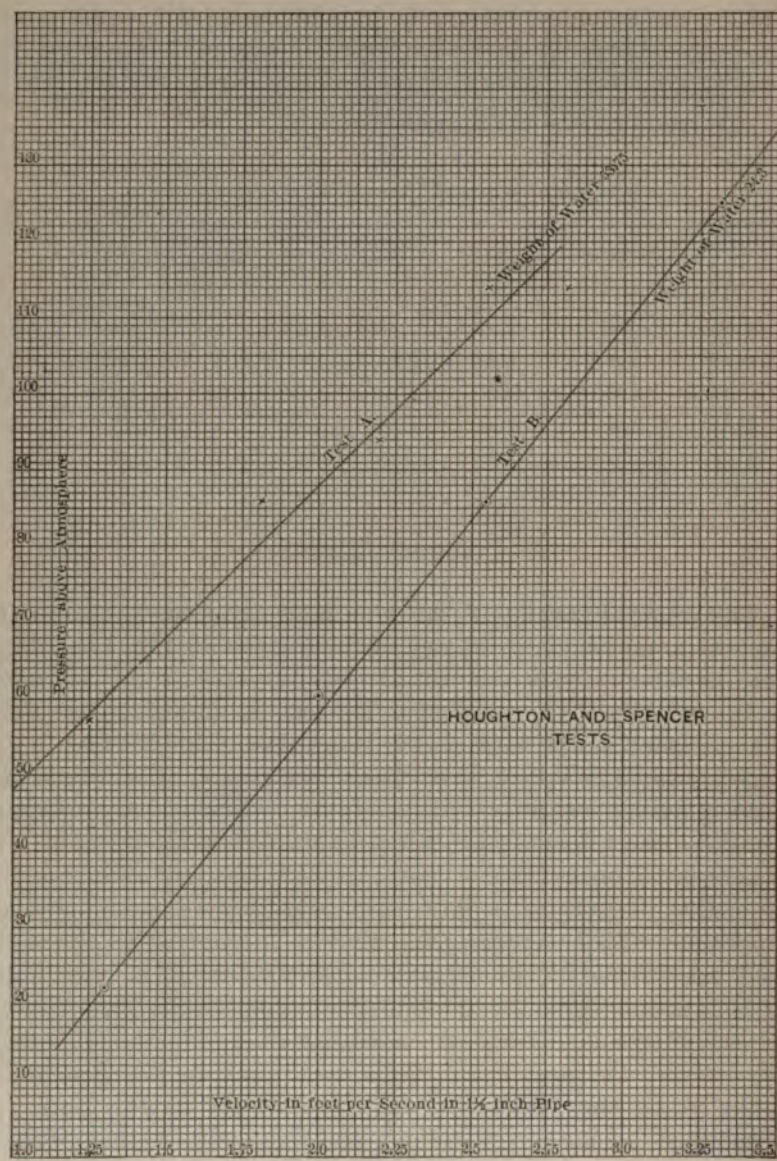


FIG. 154.

That representing the tests made in case B would be

$$p' = 51\,v - 43.$$

It will be noticed that $p = p'$ when $v = 5.35$.

While other experiments are needed to make our information more definite and certain, it would seem that the effect of the water, however, is a function of, first, time of closing the valve; second, velocity of column of water; third, mass of water in motion.

Prof. De Volson Wood.—I was interested in this practical experiment, and sought data to see if I could make a formula that would be of any service in the experimenting. So far as investigated, I concluded that an experiment similar to this—I don't know about the one that is reported in the paper, in which the pipes have many angles and different diameters—but where the mass passes through a uniform or comparatively uniform tube, that the water hammer would vary nearly inversely as the time of closing the valve. By closing it exceedingly quickly, a very high pressure might be obtained. There is one difficulty in the investigation, which I have not thought out, and that is in regard to the elasticity of the water. In the investigation made I assumed that the energy of the water was all destroyed in producing the hammer. But the fact that water is elastic, is vastly more compressible than iron or steel, will modify the numerical results. It may be yet of some service in making experiments, and I am very glad to get these last or later facts in regard to it. I noticed apparently in the first figure that the closing of the valve was uniform and that the pressure rose uniformly with the time. That is one of the elements which I was looking for. Since then Professor Carpenter's assistant has informed me that I cannot depend upon the fact that the rate of closing was uniform. Still, where it is exceedingly quick, perhaps the laws deduced will be of some service, although not strictly exact.

The following is the analysis:

The work done during the closing of the valve will be

$$\int p\,ds - \int_0^{t_1} p\,v\,dt \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

in which p is the pressure on the valve at time t, and v the velocity of the water through the gate. It is observed that the pressure above the normal varied nearly as the time, and assuming this to be the law, we have

$$p_1 = a\,t,$$

where a is a constant. If p' be the normal pressure, then

$$p = p' + p_1 = p' + a\,t.$$

Assuming that the velocity varies inversely as the gate opening, and if k be the area at full opening, then at time t the velocity will be

$$v = (1 - b\,t)\,V,$$

where V is the velocity in the pipe with the valve fully open. Substituting in equation and integrating gives

$$(p'\,t_1 + \tfrac{1}{2}\,a\,t_1^2 - \tfrac{1}{2}\,p'\,b\,t_1^2 - \tfrac{1}{3}\,a\,b\,t_1^3)\,V, \quad . \quad . \quad (2)$$

When the time is t_1, the pressure will be greatest, which call P; then

$$P = a\,t_1$$

and at the same time $v = o$;

$$\therefore,\ 1\,b\,t_1 = o, \text{ or } b = \frac{1}{t_1}.$$

These reduce (2) to

$$(\tfrac{1}{2}\,p'\,t_1 + \tfrac{1}{6}\,P\,t_1)\,V,$$

which will be the work done per unity of section of the pipe. If it now be assumed that the entire energy of the water in the approaching pipe be consumed in producing this work, the above will equal $\frac{1}{2}\,M\,V^2$, and if l be the length of the pipe, w the weight of unity of volume of the liquid, then for unity of section, the kinetic energy will be

$$\tfrac{1}{2}\,\frac{w}{g}\,l\,V^2 = (\tfrac{1}{2}\,p'\,t_1 + \tfrac{1}{6}\,P\,t_1)\,V;$$

$$\therefore,\ P = \frac{3\,w\,l\,V}{g\,t_1} - 3\,p'.$$

The total maximum pressure would be p' greater, or

$$P + p' = \frac{3\,w\,l\,V}{g\,t_1} - 2\,p'.$$

If the initial pressure be zero, or so small that it may be neglected, the

$$P = \frac{3\ w\ l\ V}{g\ t_1};$$

Or, the maximum pressure would increase with the momentum $\left(\frac{w\ l\ V}{g}\right)$ and inversely as the time of closing the gate.

Mr. C. W. Nason.—In the construction of a formula, if niceties are to be observed, there would also have to be taken into consideration the elastic expansion of the tube, which would probably be even a greater factor than that of the compressibility of the water, to affect it; then, if a formula were devised, the difficulty of application would lie in measuring accurately the time of valve closure or stoppage of the water column. This would evidently be only an infinitesimal part of a second, and it would require an extremely accurate apparatus to measure the interval of time between the first retardation of movement in the water and its final cessation of flow.

*Prof. R. C. Carpenter.**—The formula given by Professor Wood seems to agree fairly well with the results of experiment A, but does not give a uniform coefficient for experiment B. It was not possible to determine with exactness the time of closing the valve, but it seems quite probable that in the various experiments the time of closing the valve was fairly uniform, and hence we can consider this quantity a constant.

His formulæ for our experiments, supposing valve closed uniformly, would be reduced to the form

Maximum pressure = *constant* × *weight* × *velocity* (C).

In the following table each coefficient is given.

Experiment *A*: *wt* = 53.75 lbs., head 4 ft.				Experiment *B*: *wt* = 24.3 lbs., head 4 ft.			
No.	Velocity.	Max. Pressure.	Constant in Eq. C.	No.	Velocity.	Max. Pressure.	Constant in Eq. C.
1	1.25	67.2	0.86	1	1.29	22.5	0.72
2	1.82	98.	0.88	2	2.01	61.	1.25
3	2.2	118.5	0.802	3	2.54	85.	1.39
4	2.55	137.	0.83	4	3.32	125.	1.56
5	2.85	153.5	0.745				

* Author's Closure, under the Rules.

It is noticed that for experiments A the coefficient is fairly uniform, but for experiments B they increase nearly with the velocity.

Elastic Theory.

It being well known that water is a very elastic medium and also very compressible as compared with iron or steel. it occurred to the writer that it might be of interest to consider the problem of maximum pressure which might be produced by one elastic body, *i. e.*, the water, impinging upon another body, the pipe, which is elastic but also immovable. The computation for such a case is given; the results would indicate that the maximum velocity was nearly independent of the mass and dependent only on the velocity. It is to be noted that this supposition requires the mass of moving water to be instantly stopped, or that the time required for closing the valve be zero. It is of course an approximation, but it is of interest, especially since it is the only hypothesis that will explain fully the recurrence of wave pulsations at equal intervals of time.

Suppose that $M = \frac{G}{g}$ impinges with velocity, v, upon an immovable mass, all its *vis viva* is expended in compressing the latter body.

Denote the maximum pressure in excess of static by P_1; the longitudinal compression of the water by λ_1, that of the iron by λ_2; the area of cross section by F_1 and F_2; moduli of elasticity by E_1 and E_2; *wt* of a cubic foot by w; total weight of water moving by G; length by l_1 and l_2; total space moved through by σ. We have for the work done;

(1) $\frac{1}{2} P\sigma = \frac{1}{2} Mv^2 = \frac{1}{2} v^2 \frac{G}{g}$.

(2) Also, $\sigma = \lambda_1 + \lambda_2$.

(3) $E = \frac{Pl}{F\lambda}$, by definition modulus elasticity.

(4) $\lambda_1 = \frac{Pl_1}{F_1E_1}$; put $H_1 = \frac{F_1E_1}{l_1}$ and $H_2 = \frac{F_2E_2}{l_2}$;

then (5) $\lambda_1 = \frac{P}{H_1}$, and similarly $\lambda_2 = \frac{P}{H_2}$.

Substitute values of λ_1 and λ_2 in (2).

$$(5)\quad \sigma = \lambda_1 + \lambda_2 = P\left(\frac{l_1}{F_1E_1} + \frac{l_2}{F_2E_2}\right)$$
$$= P\left(\frac{1}{H_1} + \frac{1}{H_2}\right) = P\left(\frac{H_1 + H_2}{H_1H_2}\right)$$

From which

$$(6)\quad P = \sigma\left(\frac{H_1H_2}{H_1 + H_2}\right);$$ substitute this value in (1).

$$(7)\quad \frac{H_1H_2}{H_1 + H_2}\sigma^2 = v^2\frac{G}{g}.$$

From which we find

$$(8)\quad \sigma = v\sqrt{\frac{H_1 + H_2}{H_1H_2}\cdot\frac{G}{g}};$$ substitute this value in (6).

$$(9)\quad P = v\sqrt{\frac{H_1H_2}{H_1 + H_2}\cdot\frac{G}{g}} = v\sqrt{\frac{F_1E_1 \,.\, F_2E_2}{F_1E_1l_2 + F_2E_2l_1}\cdot\frac{G}{g}}.$$

The equation as given above is general, but for the case of water impinging against an obstruction of the full size of the pipe, $l = l_2$, $F_1 = F_2$.

$$(10)\quad P = v\sqrt{\frac{FE_1E_2}{l(E_1 + E_2)}\cdot\frac{G}{g}}.$$

For water $E_1 = 300{,}000$ lbs.; for wrought iron, $E_2 = 25{,}000{,}000$ lbs.; for cast iron, $E = 12{,}000{,}000$.

By substitution we have for maximum pressure produced by water hammer in a wrought-iron pipe,*

$$(11)\quad P = 544\ v\ k\sqrt{\frac{F}{l}\frac{G}{g}},$$ in which k is a coefficient to be determined by trial; in which G = the total weight of water flowing, and g = the force of gravity (32 16).

In a cast-iron pipe

* In these formulæ we have supposed the diametral expansion to be zero; since the water changes in volume from forty to eighty times as much as the pipe, this error cannot be of great importance. We have used in the examples values of velocity for 2-inch pipe. The diagram gives velocity for 1½" pipe.

(12) $P = 542\ v\ k \sqrt{\frac{F}{l}\frac{G}{g}}$.

Application.—In the experiments made by Houghton and Spencer, first set (A), $l = 57.92$ feet $= 695$ inches; $F = 3.35$ square inches.

The weight of water in the pipe was 54 lbs. nearly.

Material of pipe, wrought-iron.

$$\sqrt{\frac{F}{l} \cdot \frac{G}{g}} = \sqrt{\frac{3.35}{695} \times \frac{54}{32.2}} = \sqrt{.00808} = .09.$$

So that for this case the maximum pressure

$$P = 49\ v\ k.$$

For second set of experiments (B), $l = 24.62$ feet $= 295.5$ inches, $G = 24$ lbs., $F = 3.35$.

$$\sqrt{\frac{F}{l}\frac{G}{g}} = \sqrt{\frac{3.35}{295.5} \cdot \frac{24}{32}} = \sqrt{.00855} = .0925.$$

For this case, $P = 50.2\ k\ v$.

In any case, by consulting equation (12) it is seen that since w and g are constants, substituting k' for $k\sqrt{\frac{w}{g}}$, we have $P = k'vF$ (13).

This would imply that the maximum pressure was a function of product of velocity and area of cross section of pipe.

For the cases above considered, area of pipe was constant; hence, according to this theory, for this case P would be a function of v, so that we might write $P = k''v$ (14).

Tabulating the results of the two sets of experiments, we find:

Experiment A: $wt = 53.75$ lbs.				Experiment B: $wt = 24.3$ lbs.			
No.	Velocity.	Max. Pressure.	Constant in Eq. 14.	No.	Velocity.	Max. Pressure.	Constant in Eq. 14.
1	1.25	67.2	53.5	1	1.29	22.5	17.4
2	1.82	98.	53.7	2	2.01	61.	30.5
3	2.2	118.5	54.0	3	2.54	85.	33.5
4	2.55	137.0	53.7	4	3.32	125.	37.5
5	2.85	153.5	53.7				

It would seem, on the whole, that considerable more experimental work is needed to determine the probable laws of variation due to changes of conditions in the piping or in opening or closing the valves. It is the intention to continue the series of experiments which have been already described, and the results will form the subject of a paper at the next meeting of the Society.

DLXXVII.*

STEAM PIPING AND EFFICIENCY OF STEAM PLANTS.

BY WILLIAM A. PIKE, MINNEAPOLIS, MINN.

(Member of the Society.)

In some recent experience with steam plants, the writer has been impressed with the apparent lack of good judgment and the application of correct principles in the arrangement of the piping between boilers and engines. In many cases, every effort seems to be made to select the best engines and boilers for the particular purpose in hand, but when it comes to the question of connecting them with steam-pipes it would appear that it was thought that anything with a hole through it would do. Whether the piping is right or not, affects the economy, efficiency, and safety of a steam plant more than most engineers seem to think, and the writer believes that poorly contrived piping is in many cases as detrimental to high efficiency as the use of the wrong type of engine or boiler would be. The following are, perhaps, the points that should be most particularly looked out for in the design of a system of piping:

(1) The piping and accessories should be so arranged that practically dry steam shall always be delivered to the engine.

(2) The steam should reach the engine with very little "drop" in pressure.

(3) The piping should be so planned and valves so placed that water cannot under any circumstances collect anywhere, except in places especially arranged for that purpose.

(4) Wherever practicable, and especially where more than one engine is supplied from one battery of boilers, the piping and valves should be arranged so that in case of accident to any part, it can be shut off without shutting down the whole plant.

These requirements are not offered as new or original, but simply as covering points often neglected, and as furnishing texts for this paper. The first requirement given above, viz.: that the

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions.*

steam should reach the engine dry, is one often neglected, and the writer has in mind plants of considerable magnitude where absolutely no provision is made for drawing off condensed, or entrained, water, except through the engine itself, and the not uncommon cases of broken cylinder heads and pistons testify as to the effect of such designs. In these days of efficient pipe coverings, of separators, steam loops and traps, there is no excuse, but false economy, for allowing water to reach the engine. In connection with point No. 2, a plan for disposing of water, followed with good success, will be spoken of.

(2) Steam should reach the engine with but little drop in pressure. The effect that an improper arrangement of piping has on the pressure with which the steam reaches the engine, is probably not as well known as it should be, and the principal object of this paper is to bring it to the attention of engineers, and to show how, in one case, the evil has been remedied.

The conditions of the case used in illustration were as follows:

There were ten single acting engines, 16 x 27 x 16 inches, arranged on two sides of an engine room, and making 250 revolutions per minute. They were supplied by a number of boilers, and the steam from these first passed to a drum, 28 inches in diameter and 45 feet long, at the top of the boiler room and between two rows of boilers. From this drum separate 8-inch pipes ran to each of the five engines on the side nearest the boilers. (At the time referred to, the other five were not connected.) The average length of each 8-inch pipe was about 76 feet, and there were five right-angle turns and two valves in each pipe.

On testing these engines it was found that there was a great difference between the boiler pressure and that at the engines.

This difference varied from eight to twenty pounds, depending on the amount of work being done, and averaged fourteen pounds, during usual work.

To find out, if possible, something as to the reason for this drop, an indicator was placed in connection with one of these 8-inch feed-pipes, just outside of the throttle-valve, which was close to the engine. At the same time cards were taken from the engine, and Fig. 155 represents the result.

The narrow diagram at the top is that taken from the pipe, while the lower one is a simultaneous diagram from the high-pressure cylinder.

As will be seen, the steam is at the pressure *a* at the beginning

of the stroke, and at admission falls suddenly to *b*, then during admission keeps nearly constant until complete cut-off at *c*, then the pressure rises to *d*, and gradually settles down to *a* again at the beginning of a new revolution. From these diagrams it can be seen that, as there is practically no steam-chest to these engines, and as they are single acting and cutting off at about half stroke, there is a sudden and great demand on the 8-inch pipe during admission, a demand so great that the steam cannot keep up its pressure; and, more than this, it appears that a vibration of the steam is set up in the pipe, causing the pressure to fluctuate from considerably below to somewhat above boiler pressure. This motion of the steam, of course, requires energy to be used up which is not therefore available for work in the engine.

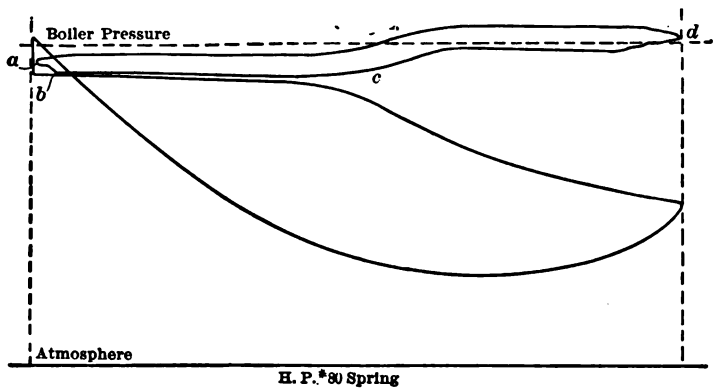


FIG. 155.

In attempting to account for this lost energy, the upper diagram has been treated as an ordinary indicator diagram, representing the work done in an 8-inch cylinder (the size of the pipe), and of a length equal to that of the volume of steam (in an 8-inch cylinder or pipe) required to supply the engine. Then, also, the loss due to friction is taken into account in the same way. The sum of these results is then compared with the extra work the steam would have done if admitted to the engine at boiler pressure. This last is shown in Fig. 156, in which the shaded part *AB* represents the additional work that would have been done if the steam had been at boiler pressure instead of having the admission line *a-b*, as was actually the case.

As an example, is taken the case of two diagrams, taken simultaneously on October 21, 1892, at 9.20 A.M.:

The area of the pipe diagram is 0.4 square inch; the length of the pipe diagram is 3.18 square inches; the mean effective pressure is 10.06 lbs.; and the length of an 8-inch cylinder, corresponding to the amount of steam used, is 3.0 feet. Then the work done = $Pe\ LAN = 10.06 \times 30 \times 50.26 \times 1 = 1{,}517$ ft.-lbs. The pressure loss, due to friction, is estimated, from the tables in "Steam," to be 2.19 lbs.; therefore the lost work due to friction $= 2.19 \times 50.06 \times 3.2 \times 1 = 330$ ft.-lbs., or a total of lost work $= 1{,}847$ ft.-lbs. The area added to the engine diagram, corresponding to AB, Fig. 156, = .28 square inch; the length of the whole diagram = 3.3 inches, and the lost mean effective pressure = 6.79 lbs.

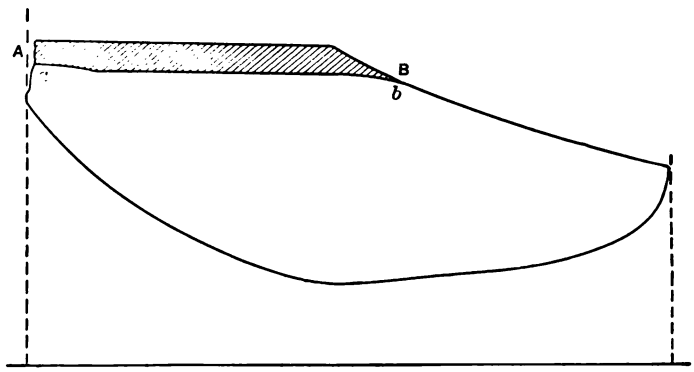


FIG. 156.

This pressure corresponds to 1,819 ft.-lbs. per stroke. Thus we find from the pipe diagram and the estimated friction a loss of 1,847 ft.-lbs., and the corresponding loss as figured by the engine diagram, 1,819 ft.-lbs., or an agreement within $1\frac{6}{10}$%.

This analysis is added as an attempt to account for the loss, and, whether correct or not, does not affect the fact that in this method of piping there is a serious loss of pressure, causing a corresponding decrease in efficiency.

As in this case, the "drop" was so great, it was considered desirable to endeavor to prevent it.

Two plans were suggested, as follows: First, when the amount of steam used was considered, it was seen that, due to the fact that the engines were single-acting, the demand while it lasted (with a cut-off of $\frac{1}{2}$, say $\frac{1}{4}$ of the time of a revolution) was excessive for

an 8-inch pipe, though all available data would show the pipe amply large for a uniform flow of that amount; and so the first idea was to enlarge the pipe, still piping separately to each engine, with the idea of reducing the effect of break-downs. But the same reasoning led to the fact that an 8-inch pipe would be amply large if the flow could only be made practically uniform. This led to the second plan—viz., to place a reservoir for the steam close to each engine, and thus enable the flow through the 8-inch pipe to be more nearly uniform. In order to get information as to such a plan, the arrangement shown in Fig. 157 was extemporized.

The 8-inch pipe leading to engine No. 2 was cut, and a drum,

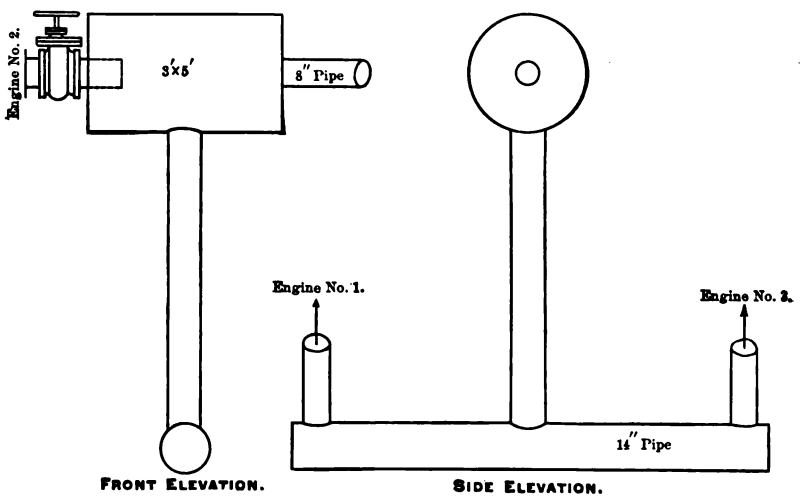


FIG. 157.

3 feet × 5 feet, put in close to the engine; from this drum a 10-inch pipe led to a 14-inch pipe, about 13 feet long on the floor, and from this 8-inch pipes led to engines 1, 2, 3. The object was to get the effect, on drop of pressure, of receivers fed by one 8-inch pipe when supplying one, two, or three engines. Much to the satisfaction of the experimenters it was found that when engine No. 2 alone was supplied, through the drum, the drop was but a trifle over 1 lb.; that when two engines were fed, the drop was a little over 2 lbs.; and when all the three were supplied by one 8-inch pipe, through the receivers, the drop was about 5 lbs. These results confirmed the theory that an 8-inch pipe was easily

ample for two engines without excessive drop, and, that when supplying three engines in this way, the drop was but little more than a third of that when but one engine was fed by a single 8-inch pipe without receivers. With this data as a basis new piping was designed for the plant, as follows: an extra drum was put in the boiler-room, and both drums connected with suitable valves to a loop of 8-inch pipe, running past each of the engines on the side nearest the boiler-room, with a drum, 3 feet × 5 feet, between this pipe and each of the engines. The nipples connecting the drums with the engines project about a foot into the receivers, as shown by the dotted lines in Fig. 157, to *a*, as separators for entrained water. Also each receiver was connected with a "steam loop," to carry this water back to the boilers. On the farther side a similar plan was followed, except that the loop was made of 10-inch pipe on account of the greater distance from the boilers. All pipes and drums were covered with an asbestos air-space covering, and valves were placed between each pair of engines to allow of shutting off any part in case of a breakdown, without shutting down the rest of the plant. Tests made of this system show a drop of only 2 lbs. on the farther side of the engine-room, when all five engines were under full load and fed by one loop of 10-inch pipe (making a complete circuit from the drums back to the drums again). The combination of "steam loops" and the separator drums is the plan referred to under the heading with reference to dry steam. This has worked admirably, there being no signs of water in the engines as there had been before.

(3) The danger that may arise from having piping and valves so arranged that water can collect anywhere except in places specially provided, can be best shown by two cases within the knowledge of the writer.

A. A battery of boilers was connected with a header, as follows. From each of two boilers a short pipe was led up to a horizontal pipe branching to the right and left to another pipe leading to the header. Valves were placed close to each boiler, but none at the header. At one time two of these boilers were shut down and the valves at the boilers closed, the remainder of the battery being in service. The first thing noticed was that the calorimeter in the header showed much dryer steam than there was any reason to expect, and on investigation it was found that the branch pipe between the two boilers and the header was acting as a condenser

and separator and becoming filled with water. This was fortunately noticed in time, and the water drawn off and a circulation kept up in the pipe, after which the calorimeter showed normal moisture, and no harm was done, but if this body of water had been in any way set in motion the effect on the piping, which in this case was of cast iron, can be imagined.

B. In the second case there was a steam main leading from boilers to various pumps, engines, etc., and at a certain point a temporary pipe was led into this main from another boiler with no valve between the pipe and main. Steam was admitted to this pipe for some reason, and then after a few minutes shut off and steam let into the main. Shortly after this steam was again admitted to the temporary pipe, and very shortly a "slug" of water passed from this pipe to the main and acquiring a high velocity flew to a dead end and blew it off, causing the death of an employee. There seems no doubt that a valve placed at the intersection of the main and the temporary pipe, with a suitable "bleeder" in the pipe, would have prevented this accident, if handled properly.

(4) The necessity of arranging piping so that in case of accident only a part of the plant need be shut down needs no argument, as the loss and annoyance due to shutting down a large plant, especially one on which the public in any way depends, as in the case of a lighting or street-railway plant, is much greater than the expense involved in making such a thing impossible. The system followed in the case cited under the second head, is one that while expensive makes it impossible for any large part of the plant to need shutting down; the valves between the boilers and drums in the boiler room, combined with those in the pipes from the drums and those between the engines, making it possible to cut out almost any part without materially affecting the rest.

While, doubtless, much more could be written on this subject, it is hoped that the cases cited above will serve to call attention to the necessity for greater care in designing systems of piping, and will call out a fruitful discussion.

DISCUSSION.

Mr. William Kent.—Mr. Pike's paper is of importance in calling attention to the facts: (1) That the ordinary formulæ

for dimensions of steam-pipes are not to be relied on when the demand for steam in the engine cylinder is of such an intermittent nature as to set up vibrations in the pressure in the pipe; (2) that steam receivers placed close to the engine are of great value in such cases; and (3) that there are hidden dangers in any system of piping in which water can accumulate anywhere except in places provided for it. I would, however, take exception to a portion of his theory concerning the steam-chest diagram, which shows the pressure in the chest to be occasionally greater than that in the boiler. He says, "This motion of the steam, of course, requires energy to be used up, which is not, therefore, available for work in the engine." By the doctrine of the conservation of energy, energy cannot be used up, but only converted into other forms of energy. If the steam-chest diagram is a diagram of energy, or of work done, which can be measured in foot-pounds, we must ask, what becomes of the energy, or upon what is the work done? As work is the overcoming of resistance through space, what is the resistance and what the space? If the energy is converted into heat, it is not lost, but reappears as extra heat in steam, which may be useful in diminishing condensation in the cylinder. The fact is that the steam-chest diagram is not a diagram of energy at all, but only a diagram showing the existence of vibrations; such a diagram as a vibrating spring might make when the spring was doing no work except overcoming the resistance of the air in which it moves, and the resistance of the pencil on the diagram paper.

It is doubtful also whether the loss due to friction should be calculated as a loss of work, and measured in foot-pounds; and certainly it is not clear why the work should be calculated as the product of uniform pressure, 2.19 lbs. acting through a definite space of 3 feet. If the pressure loss is 2.19 lbs. at the end of the pipe next the engine, it is 0 at the other end, and is not uniform through the length of the pipe, or through any portion of it, as 3 feet. Moreover, if the friction loss is a loss of work, the work is converted into heat, which partially dries the steam. Of course, neglecting possible drying or superheating, the drop of pressure is detrimental, in so far as it makes it necessary to carry steam farther in the stroke and to exhaust at a higher terminal pressure for the same work than would be the case if there were no drop, and this criticism of a portion of

Mr. Pike's theory in no wise detracts from the value of the practical results shown in the paper.

Mr. A. C. Walworth.—I can illustrate some of the points in this paper by referring to a plant recently put up for the West End Street Railway in Boston. Mr. Levi R. Greene, the engineer who put the plant up, hoped to be here to-day to explain it himself, but as he is not here I will try to say something about it, though I am not as familiar with it as he is. One of the peculiarities of this plant is that it is a very large one, and another peculiarity is that it is made in duplicate, so that there is a duplicate system of pipes from every boiler to every

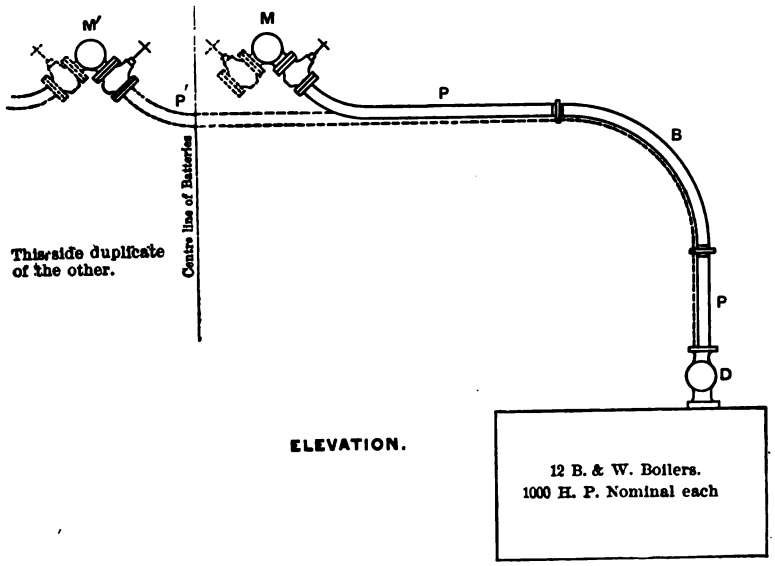


FIG. 158.

engine, and this is very carefully carried out. In general, the lay-out of the plant is like this: the seven engines are in a separate room and on both sides of the room, three or four on a side, and the steam mains simply come down the centre of the boiler space, as at *M* and *M'*, Fig. 158, with large branches to the engines and also to the boilers. In this plant, as I have said, there is a duplicate system of mains. The main *M'* is a duplicate of *M*, and each boiler is connected to each main, as shown, so that there is a complete duplication of all the steam pipes between the boilers and the engines. The boilers are of

the Babcock & Wilcox type, being arranged in batteries of three each. In the first place, the 8-inch drum which comes with the Babcock & Wilcox boiler was dispensed with, and a 30-inch drum, *D*, was placed across the top of the boiler, in order both to give a freer circulation of steam and also to act as a separator. Of course, this 30-inch drum was connected to each section of the boilers with the usual vertical pipes. Then two 8-inch pipes are run to the front of the boiler, *P*, *P'*. Thus duplicate pipes were taken from this 30-inch connecting drum over to the mains, *P* going to the nearest one, and *P'* to the further one. The group of boilers on the opposite side had the same connections, one pipe coming here to *M'* and another pipe coming to *M*. I can give by an elevation a rather better idea of the connection between the steam main and the boilers. That would represent the setting of the boilers, for instance, and in elevation. There is the 30-inch drum, *D*. Here are the duplicate steam mains, *M*, *M'*, running to the engines. This 8-inch pipe, *P*, starts from a nozzle of Midvale steel riveted to the drum, comes over in this way not with an elbow, but with a bend, *B*, made with an 8-inch charcoal-iron pipe, and strikes the bottom of that main at an angle of 30°. At this point is inserted the 8-inch main valve controlling the boiler, this conducting pipe, of course, pitching toward the boiler. Underneath there is a gallery from which these valves are handled. The duplicate pipe from this drum also goes over and comes up into the main, *M'*. The duplicate pipes from the other boiler come in from the other side with corresponding valves. There is no danger of water collecting in the pipe, *P*, but if the 8-inch valve is shut off there is a small pocket left, so that a drip pipe is taken from the body of the 8-inch valve and carried to a main drip drained by a steam trap. As there is a small water space in the large conducting main, a drip is also taken from that and from all these valves in the same way, and they are all conducted to the main drip. These valves are Chapman valves. The pressure is 160 to 180 lbs. Perhaps the most peculiar feature about this whole system of piping is that there are no cast elbows used. All the turns are made by long turns bent in the pipe. Steam-pipes as large as 20 inches in diameter were bent for this plant by Mr. Greene at Pittsburgh. Pipe as large as 20 inches in diameter was bent on a 10-foot radius without any deterioration. When you bend a steam-pipe

of any size the tendency is to flatten the section across the centre, so that the section through the centre of the bend would show an ellipse. These pipes were carefully measured after they were bent, and the difference in the two axes of the elbows of the 16-inch pipe was found to be only half an inch, so that practically there was no reduction of area in the pipe from bending. A certain compression of the metal of the pipe was found, which represented itself by regular corrugations on the inside of the pipe, showing no marks on the outside, so that the pipe was efficient—rather stronger at that point than weaker. The flanges that were used in putting the pipe together were of compressed steel, being riveted to the pipe; the flanges, also, being riveted flange to flange, the web of the flange being made wide in order to admit of a sufficient number of rivets being riveted in place. The peculiarity of this flange is that in facing the flange, instead of coming out as a duplicate of the other flange, the face was beveled off in this way and made a little smaller, making a joint where a tool could be used to calk the flanges tight all the way around, the steel flanges being riveted together in the usual way. The valves, of course, do not have steel flanges. The valves have a tongued and grooved flange, made on the valve, and a steel flange was made to correspond to that, and those flanges were bolted together.

Mr. Oberlin Smith.—Can you explain the method by which the pipes were bent?

Mr. Walworth.—The pipes were bent by Mr. L. R. Greene, and it would perhaps be for him, rather than for me, to state how.

Mr. W. B. Le Van.—How did you take care of your expansion in that system? There is no provision there for expansion that I can see.

Mr. Walworth.—Where a pipe is made with a bow as large as at *B*, that expansion takes care of itself. We will suppose that *M B* is perhaps 15 or 20 feet. The expansion is only one-third of an inch, and the spring is taken out at *B*. As the main expands, those pipes have an ample swing.

Mr. Le Van.—In my practice I do not use expansion joints in connecting up steam-pipes. The following sketch (Fig. 159) will explain how I compensate for expansion: I depend entirely on screw-joints and flanged unions for the expansion, and prefer wrought-iron pipe on account of its lightness, as well as the elas-

ticity of the material and the capability of adjustment of the screwed ends. I have had in use in one plant eight (8) 150 H.P. boilers, constructed as per above sketch, for the last five years, and there has not been any trouble with any of the steam-pipe connections. The large diameter pipe carries steam for supplying the engines, and the small pipe steam for the pumps. The pocket, formed when a valve is closed, is connected by a small drip pipe into the main steam-pipe.

Mr. Eckley B. Coxe.—I do not know that you are aware that running in a coal mine is in many cases what you might call peripatetic engineering — that is to say, you are constantly moving your pumps and engines. We used to move our boilers a good deal, but we don't now. At one of our collieries (Eckley), which we took hold of in 1876, the waterworks supplying the colliery with fresh water were at 4,600 feet from the colliery. There was a cheap boiler there, and we hauled the coal over a mile, and the boiler plant was such that we could burn nothing but pretty high-priced coal, while at the colliery we were burning coal worth twenty-five or thirty cents a ton. We first figured on a compressed-air plant. We found that that involved con-

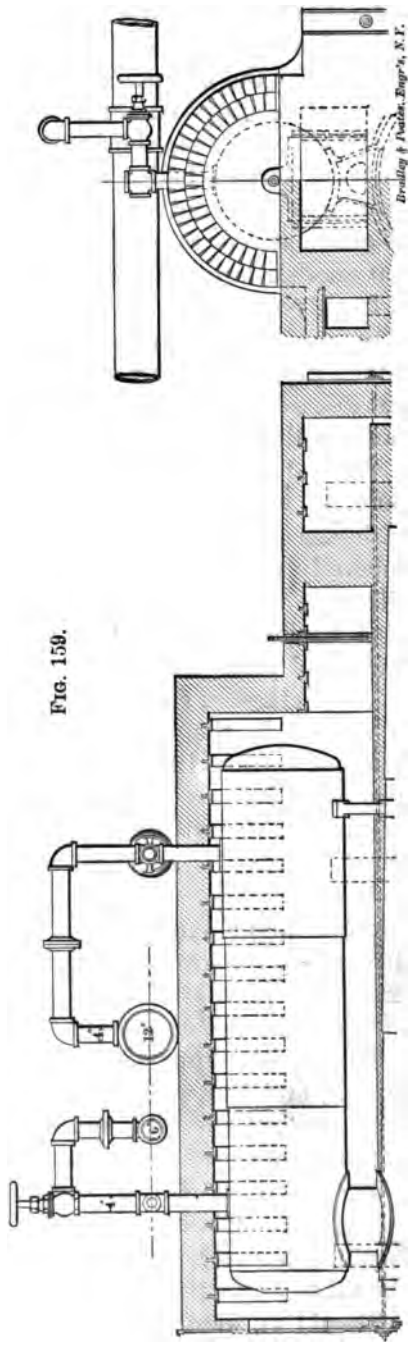


Fig. 159.

siderable loss of power, and we then took up (having had an extensive experience in the matter) the idea of carrying our steam over the mountain to the pumps—4,600 feet.

At the colliery the coal went directly to the boilers. The springs in the neighborhood had been dried up by the mine working, but we had a very fine spring over the mountain, not quite one mile south of the colliery; and by arranging matters as above stated we have been able to supply not only this one, but another large colliery, and it is done very economically.

In this case we carry the steam through a 3-inch pipe, but in other portions of our works we carry it in 2, 3, and 4, and even 6-inch pipes in exactly the same manner. In fact, we have just put down a 6-inch pipe to carry steam nearly half a mile at the same colliery.

We had two sets of boilers, one of which had more capacity than we needed, and the other had less; we connected them with the 6-inch steam-pipe and got very good results. We generally use what is ordinarily known as gas or steam pipe, although this 6-inch pipe is what is known as "spiral weld" pipe, having bought it after the company stopped making it, at a reduced price, and it is working very satisfactorily.

The method of covering the pipe is the same in all cases, as is shown in the accompanying sketch (Fig. 164). We use hemlock boards, 12 to 14, and 16 feet long, and of proper width. The troughs are made in the shop by nailing two boards together and putting triangular strips in the bottom, and another trough is made to cover over this with the triangular strips in the top. It is so arranged that the upper trough is slightly larger than the lower, so as to cover it. Stakes are driven into the ground every 6 to 8 or 10 feet, according to the size of the pipe, and the troughs are laid in the angle formed by the stakes, so as to keep the pipe above the ground.

The pipe is supported on wooden or iron strips placed in the trough, and the whole is then filled with mineral wool and the cover put on. It is not necessary to pack the mineral wool, but, on the contrary, it is better to have it rather loose.

The stakes are not all the same height, but are arranged to give a more or less even grade to the pipe. Of course, arrangements must be made to drain out the water at the low point should the pipe pass over a valley.

Should we want to change the location of a line of pipe, we

take off the top, put the mineral wool into bags, move the pipe, pull up the stakes, and put them down where we desire.

Of course it is necessary to provide for expansion and contraction of the pipe. We seldom use expansion joints, but generally put "S" bends in the pipe, and if the pipe makes a turn in a vertical or horizontal direction, this turn acts as an expansion joint; but where there are no such turns it is necessary to so place the line of pipe as to bring in these "S" bends from time to time. We run sometimes 1,000 feet or more with one of these bends, depending, of course, upon the local conditions.

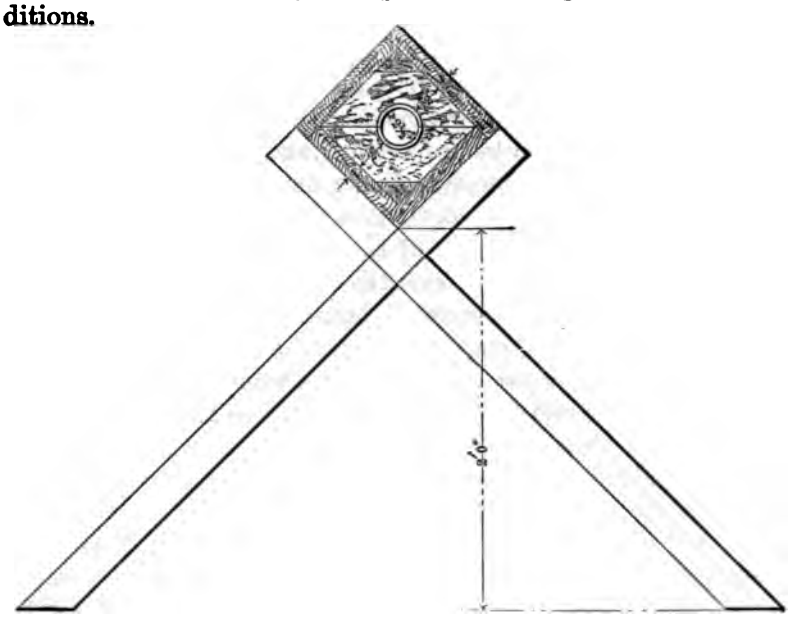


FIG. 164.

The "S" bends are made by bending the pipe at a radius of from 5 to 10 feet, according to the size of the pipe. In the beginning we filled the pipe with sand in bending it, but now the men often do it without the sand, and have become quite expert in making these bends. Of course the bends have to be inclosed in mineral wool in such a way as to leave room for them to move without breaking the boxes, but this is comparatively simple.

The line above referred to has been working since 1877; and we have not been obliged to spend anything for repairs, and have had no trouble whatever with it.

The outside of the box does not get very much heated. In winter, when a light snow is drifting, it will remain on the top of the box without melting, thus showing how very little heat is being radiated.

The great secret of success in running these long lines of pipe is never to put your steam into the engine or pump directly. There should always be placed close to the engine or pump, and between them and the main line of pipe, a large receiver, into which the steam is carried, and from which the engine gets its steam. The steam should come into the bottom of the receiver and go out at the top, the water formed being drained out from the bottom of the receiver, although there is not much water formed when the line is properly constructed.

When we have 100 lbs. pressure in the boilers, and carry steam 4,600 feet, we only have two or three pounds less at the receiver. The advantage of the receiver is not only that it takes out the water, but that it allows a continuous flow of steam, instead of getting it in at every stroke of the engine. When the engine takes one cylinder of steam from the receiver it does not diminish the pressure in the receiver appreciably, and the flow of steam from the boiler continues nearly constant. We always put receivers at our pumps and engines when the steam is carried any distance, and find them very effective and useful.

The receiver should, of course, be incased in mineral wool.

Another advantage of the receiver is that you avoid getting water into the cylinders of the pump or engine.

Mr. Oberlin Smith.—What is the amount of loss of steam or pressure?

Mr. Coxe.—I really do not know exactly. It does not amount to much. The question is, Is it cheaper to pay five dollars a day to haul coal over a mountain and pay two men to fire, or to waste say two, three, or four per cent. of your steam, which does not cost much? I do not say that a man who is paying five dollars or ten dollars a ton for his coal ought to do as we do. The total cost of the steam is not one-half what it would cost to cart the coal to the boiler. But the point in this case is that we can take steam anywhere and almost any distance. I would not be afraid to undertake to carry it nearly two miles, and it is a cold country, which lies nearly 1,800 feet above the sea, and we have about nine months of winter and about three months late in the fall. It is not great engineering, but what I call it is peripatetic

engineering; and you will make more dividends if you do it that way than in any other.

A Member.—Won't you please tell us the size of the reservoir you employ there?

Mr. Coxe.—I cannot tell you that exactly. Our standard boilers, of which we had a great many, are thirty-four inches in diameter, and we generally make the receiver of two, three, four, or five rings of those old boilers, and we cut off a portion of the boiler with the head in one end, and put a blank head in the other.

A Member.—Holding a hundred cylinders full of steam, or more?

Mr. Coxe.—Twenty-five or more.

A Member.—How often do you put your expansion?

Mr. Coxe.—We generally put them where we have a change of direction. In this case we have two.

A Member.—How far apart?

Mr. Coxe.—These expansion joints, I imagine, were considerably over 1,000 feet apart. I used to put in sliding expansion joints, but they are generally like a great many men in this world—they don't work. In most cases I do not think there was any part of the whole system that expanded as little as the expansion joints. But when you have a bent pipe it is bound to expand, and if it breaks, it doesn't break you.

Mr. Oberlin Smith.—What is the minimum thickness of the wool around the pipe?

Mr. Coxe.—It depends on the size of the pipe. We run up from 2½ to 4 inches. Good results are obtained with 450 lbs. per 100 feet for 3-inch pipe. Of course, if you were to make a round box you would not take as much mineral wool, but it would cost you more for the box. It is only a question of cost. We buy the mineral wool by the car-load.

Mr. W. A. Pearson, Jr.—I think this subject, as Brother Pike has said, is an important one—one which I think most of our mechanical engineers have not considered. When our friend Mr. Coxe speaks of his piping here, he has not quoted the pressure that he is using.

Mr. Coxe.—I can give it to you. We are carrying about 100 lbs. on that line.

Mr. Pearson.—It has been my pleasure, during the past year, to renovate the high-pressure steam piping in the plant of

the General Electric Company at Schenectady. We have two systems there: what we call a "low-pressure" system and a "high-pressure" system. The low-pressure system is of the old type, and we have had some slight trouble in keeping it tight; but we had fully 500% more trouble with our 160 lbs. pressure plant before we renovated it, than we did with our 120-lbs. pressure plant, thus showing the great difficulty of housing and taking care of steam as the pressure increases. We had wrought-iron piping with cast-iron flanges and packed joints. The consequence was we had a gang of men constantly employed Saturday night and Sunday repairing joints. We would sometimes have a joint blow out, and it could not be repaired until Saturday night or Sunday, as we are compelled to keep the steam on the mains from Monday morning until Saturday night, making it very expensive to us in the amount of steam that escapes, besides the annoyance and damage to our dynamos.

I first laid out for extra heavy wrought-iron piping with cast-iron fittings and steel flanges. After laying it out I was not satisfied with the plans. After considering the matter, I came to the conclusion that if I stayed at home and did not see what others were doing in the same line, I could not expect to obtain the best results; so I went to Boston, New York, Brooklyn, and several other places, and visited all their principal plants, and in the whole course of my journey I failed to find a single plant that was not troubled with escaping steam where they were carrying high pressure. Therefore, when I returned home I destroyed the plans already made, and decided that I must use cast-iron pipes. The reason why I came to this conclusion was:

First.—If cast iron was good enough for valves it certainly must be all right for plain piping, as it was a much simpler casting to make perfect.

Second.—I concluded that by using cast iron I could cut the number of joints necessary down to about one-quarter of what would be required by the use of any other metal.

Speaking with some men, they would say, we have only one joint in 50 or 100 feet of wrought-iron riveted piping. In this connection I would say that each rivet and seam are separate and distinct joints, and will give more or less trouble to keep tight if you deal with high pressures.

We have cast-iron piping throughout our high-pressure plant, with the exception of some copper. When I put in this copper I did not have an expansion joint which I have invented since that time, and which I believe Professor Thurston saw tested in the Pearl Street station. Since then I have dropped the use of copper and have used cast iron exclusively, with the exception of a steel separator, which is 4 feet in diameter and 6 feet high. Westinghouse, Church, Kerr & Co. submitted plans showing steel ⅜ of an inch thick. I doubled this thickness. They were anxious to know why I increased the thickness so much, and I stated that I did not want to take any chances on its not being tight. So we therefore put in a separator of ¾-inch steel, and it was tested in the shops of the Westinghouse people at Schenectady at 500 lbs. hydraulic pressure, and did not show a particle of leak anywhere. We put it in place, and in the course of the last eight months we have been compelled to calk it in a hundred different places. This forced me to the conclusion that we did not want any leaky rivets or seams. The only solution of the problem was to use throughout cast-iron piping and ground joints. I put in a separator a little farther on in the system, made of cast iron. This was put in operation eight months ago, and in that time we have not seen a particle of escaping steam from any part of the whole system excepting what we have seen come from the steel separator. I think two of the most important points in steam piping are water and expansion. In erecting steam piping according to the method shown on the blackboard, connecting into the main at an angle of 45°, there is quite a pocket formed there. Those of you who are familiar with all valves know that each valve has a water pocket in it. There would also be a water pocket from the face of the valve to the pipe.

After going over the whole matter carefully I concluded that the only way to connect into the pipe was at the side or top, as it is very important that no water be allowed to accumulate in the pockets or separator of the system. If it gets started it forms in the shape of a ball and goes through the system like a cannon ball, which is liable to cause very serious accidents. They had an example of this at the World's Fair; it blew out a blank flange at the end of a main pipe, killed three men, and displaced some very large timbers. In our plant at Schenectady we have endeavored to take care of every particle of water in

the whole system. We have the loop system from every valve, or placed wherever there is the slightest water pocket in the system. After going over the trap and loop systems thoroughly I concluded that the loop system was the best, and it has worked very well since it was installed.

Next comes expansion: In figuring the expansion on cast-iron piping all the engineers I have talked with claim that one inch to 100 feet is the largest amount they had ever found to occur in cast-iron piping. In a length of 150 feet we got 3 inches expansion. All engineers here to-day will agree with me on this point, that when the expansion occurs in steam piping it comes by jerks, and the cause of this is the internal strain. There is no way to deal with it outside of this expansion joint of mine. You will agree with me that what engineers have had to contend with was the area of all sizes of piping, and when it comes to a large pipe of 16, 18, or 20 inches, and in some cases 24-inch mains, the end strain is enormous. In a plant in this city to-day they are putting in piping to carry 250 lbs. pressure, and I believe they are going to put in 22 or 24-inch mains. When we figure the area at that pressure we can see at once the enormous end strain if they use the old slip joint. They may use the copper "L," which is the only remedy engineers have been able to find heretofore, with the exception of the old slip joint, male and female. Using this, both ends of the pipe have to be blocked to withstand the enormous resistance. With this joint of mine you can erect a system of steam piping, and it matters not how long or how extensive the system is, you can take care of every particle of expansive strain. In the main of 150 feet of piping, which I heretofore referred to, there was 3 inches expansion, and there was no jerking, as the expansion came on gradually. As we all know, there is nothing that will heat iron as quickly as hot water or steam. This is the reason I was converted to cast-iron piping. If any of you gentlemen here to-day were getting up specifications for wrought-iron steam piping you would all specify that the flanges be screwed on tight, or, if riveted, they were to be made tight, and the consequence would be that if the flanges were heavier than the pipe the pipe would have to yield and the flanges would still hold their size. I found in cases in this city, where cast-iron flanges were used, that many of them cracked. Where they used steel flanges the pipe reduced, and consequently there were leaky

joints, so I concluded that cast-iron piping with ground joints would obviate all this trouble. We have a way of grinding the joints which makes them come very inexpensive. Consequently we had, in a 12-foot length of piping, only one single joint for any trouble to arise from. The reason why cast iron was dropped out of use for steam piping was this: when it was used, they did not make the body of piping or flanges heavy enough.

If you will recollect the standard flanges and body that were prescribed to us one year ago, you will remember that the thickness of the flange for a 14-inch pipe was $1\frac{3}{8}$ inches; thickness of body, $\frac{3}{4}$ to $\frac{7}{8}$ inch; size of bolt, $\frac{7}{8}$ to 1 inch; number of bolts in flange, 12. When you compare these figures with the sizes I used on the same size of pipe, that is, thickness of flange, $2\frac{1}{2}$ inches; thickness of body, $1\frac{1}{4}$ inches; size of bolt, $1\frac{1}{4}$ inches; and number of bolts, 20, you will at once realize that there is no possible chance for the joints we use to give out. An engineer in going over these figures may at first say: this is very expensive piping. When he looks still farther he will find that the extra metal will not cost more than twenty dollars per ton (as the labor is just as great in either case), which cuts a very slight figure in the total cost of a system of steam piping.

Mr. Coxe.—Of course, as I said before, this work was not put up in New York city, where people do not want their spectacles covered with moisture. It is put up in the woods. It is peripatetic engineering. If you made a plan for it, before you had your plan completed your practical man would have it up and in running order.

Expansion joints must be used even in our domestic life, but you must put them where nobody will see them except the fellow who has to keep them in order. You do not want them to go off when you have friends to dinner. We do not use, practically, any elbows. Whenever we go around a corner we bend the pipe around, the radius of the curve being pretty large.

Mr. G. M. Sinclair.—A method used some time ago in the design of a large steam pipe may interest the Society. It is based directly on the "commercial efficiency" of the pipe, to use a phrase perhaps not heard often enough. The question the engineer has to answer is, "What, considering all circumstances, immediate and remote, will give the greatest return for the

outlay?" From this point of view it is seen there are three things affecting the cost of operating a steam pipe, viz.:

1. Interest and depreciation.
2. Condensation.
3. Friction.

The larger the pipe the greater will be the expense due to the first two, and the less the expense due to the last.

To illustrate the effect of these various items of expense I append a table in which the daily expense from each source is shown for a number of sizes of pipe. I regret that I have not the figures of the original case, but the case I have assumed illustrates the subject, although I have eliminated complications and refinements which might claim attention in practice. The problem I have assumed is the transmission of 50,000 lbs. of steam per hour a distance of 1,000 feet, with 100 lbs. initial pressure. I counted upon three right-angled turns and a globe valve. Following is the table referred to, showing the cost per day, itemized and in totals.

Diameter of Pipe.	6 in.	7 in.	8 in.	10 in.	12 in.
1. Interest, etc.	$0.39	$0.46	$0.53	$0.66	$0.84
2. Condensation	1.51	1.76	2.01	2.51	3.02
3. Friction	0.86	0.38	0.19	.06	.02
Total	$2.76	$2.60	$2.73	$3.23	$3.88

As to the question of how the above costs were obtained, I can only outline the principles involved. Particular cases require particular treatment. To get the first item of expense, a careful estimate must be made of the cost of the pipe, including all fittings, coverings, hangers, labor, etc. The necessary outlay of capital being determined, the usual interest, say 6% per annum, must first be allowed. In addition, we must allow a percentage to cover depreciation and repairs. This additional percentage must be sufficiently large to put in hand the value of the pipe, when it becomes necessary to renew the same. It may be regarded as a sort of sinking fund. In the above case I have assumed a total of 12% per annum.

Condensation I have calculated as varying directly with the inside service of the pipe, and the friction I have figured on the formula referred to by Mr. Pike. From these calculations the loss in heat units is readily obtained. If, now, a certain

number of heat units have been produced in the boilers, and are lost in transmission, it is only necessary to know the cost of producing those heat units. This is most readily obtained by basing it on the amount of coal burned in producing the lost heat units, and this involves the cost of the coal, the labor in handling it, and interest, depreciation, etc., on the boiler plant. In the above illustration I adopted $\frac{4}{10}$ of a cent per pound of coal. While elaborate calculations, such as are indicated above, are not always helpful, there are many cases where I believe this method would prove a most valuable check.

Mr. Daniel Ashworth.—The subject of steam piping, from the popular stand-point, seems to have been looked upon as secondary, and as scarcely worthy of consideration, for a long period. In rolling-mill practice, in which I have closely followed this matter, I have had many vital defects, as regards transmission of power, through injudicious steam piping, and principally from the lack of steam-pipe capacity. That can be attributed, in the first place, to the good old days when the wheels would revolve slowly and the loads were light; and again, to the fact that the plant started with the view that that was to be the end of it—no idea of growth; but in the course of time additions were made to far outgrow the capacity of the steam distribution; and in the next place the remarkable faculty, the inborn intuition, of the steam-pipe fitter to marvelously arrive at a given point with the greatest amount of bends, knuckles, and turns that are conceivable to the human mind. I have looked at steam-engines fitted up with just such arrangements, and I have wondered how much time they spent in the drawing room in planning it, in order to get around the angles and bends. I found, some ten years ago, that some reputable steam-engine builders entirely ignored piston speed, one of the most important matters in this matter of steam distribution through our pipes. That brings us back to the first point I made. This was all well and good when the piston speed was slow and the loads were light, but as we have increased these things, this piston speed becomes one of the important factors. Referring to this matter of the steam receivers as reservoirs, etc., I would say that it is the practice throughout the whole Carnegie system, upon any engine that has to perform anything like heavy duty, always to put in receivers. That is a matter that the late Captain Jones was exceedingly proud of. The doubt that arose in my mind was

prompted by an inquiry put by Mr. Oberlin Smith as to the amount of loss by condensation, and the first thing the captain would do would be to open the cock and show how dry the steam was. We consider those receivers very important. I know by professional experience, that when I received a letter about some trouble among the rolling-mills in the Ohio valley, I just pictured up in my mind that there is a radical defect in the steam piping. The records show that nine-tenths of the trouble has been improper steam piping in those plants which have been in existence probably twenty and twenty-five years, and it would surprise you, the enormous amount of work that has been done in ten years of reforming, tearing down, and putting up proper steam-pipes of liberal capacity. Many a plant has been lifted out of the Slough of Despond by giving it proper steam piping of adequate size.

Mr. W. A. Pearson.—We have some slight trouble in keeping these different mains free from leaks at Schenectady, in our two or three miles of steam piping, varying in size from 1 to 8 inches, carrying 120 lbs. pressure. We also have, right in the same power station, a system carrying 160 lbs. pressure, and, as stated before, we had a great deal of trouble in keeping this system free from leaks. However, since we renewed this system of piping we have not had any leaks, and it has been in operation eight months. This goes to show that our present method is the best possible one for high-pressure systems. I carried out the method Mr. Taylor suggested, bringing the joint out to the inner side of the bolt hole, and in this way we have had no trouble whatever with our joints.

In speaking about the expansion-joint question, we have, right where any man can see them when he enters the power station at Schenectady, two 12-inch joints. One of them has been in operation eight months, and in that time it has required no care whatever, and steam has been on it constantly. The take-up is one inch.

I examined one plant in New York city where the steam piping cost $20,000 to put in, and the engineer in charge told me it cost $60,000 to repair it. This was wrought-iron pipe.

I visited another station where the piping cost $5,000 to put in, and the engineer in charge stated that it cost $15,000 to repair it. This was also wrought-iron pipe.

Both of the above stations were carrying 160 to 165 lbs.

pressure. We are all advancing in the line of raising our steam pressure, and this is my reason for saying what I have said here. Every engineer that I have shown this expansion joint to says it is something they have all been looking for and have just found.

There is one other point I would like to speak about, and that is, the covering of steam piping. I tested this matter very thoroughly, and would give the following results:

We had the asbestos people and the magnesia people put on samples of their material. They all said they could not carry out our specifications. These specifications stated that the temperature of the outside of the covering should not exceed 2° Fahr. above the temperature of the room. It could not be carried out in one sense, but we tried it in a temperature of 110° Fahr., and we found the temperature of different materials to be as follows: we first hung wood in the same line of atmosphere as the piping, and as near as I can recollect the temperature was 142° Fahr. We then hung magnesia blocks in the same atmosphere and found it was 140° Fahr. We tried canvas, and found it was 138° Fahr. With hair felt it was 135° Fahr.

The two coverings we had put on were magnesia and asbestos. There was one other covering, the mixture of which I cannot state, but this was not considered. We tried from 1 inch up to $2\frac{3}{4}$ inches of the two different coverings. We found that one layer of magnesia blocks $1\frac{3}{8}$ inches thick, then one layer 1 inch thick on top of the first, and $\frac{3}{8}$ inch of magnesia plaster next, and two layers of canvas on the outside, made a perfect covering, as the thermometer showed the temperature next to the canvas on the outside of the covering to be the same as that of a piece of canvas suspended in the same line of atmosphere, thus showing that we held every unit of heat generated from the pipe in the covering, and did not let the circulation in the room have any effect on the steam in the pipe. I give these figures, gentlemen, as they may be of interest to you.

In testing we found the two materials of the same thickness compared very favorably, as the variation was only about 2° to 5° Fahr. in favor of the magnesia.

Mr. Walworth.—Was there no condensation in the pipe?

Mr. Pearson.—I do not take that stand. There will certainly be a slight condensation in piping covered in this way, caused by the radiation of the heat through the pipe to the covering.

With the covering I have described, the best possible results are obtained. If the covering was 10 inches thick the amount condensation would be the same.

Prof. R. H. Thurston.—Every one will recognize the fact that the two and sufficient principles to be adhered to in designing lines of steam piping are, first, to provide for contraction and expansion; and, secondly, to provide against standing water anywhere in the line of the outside or inside. If the pipe can be arranged so that the expansion and contraction can take place without causing stress on the material, and if it can be kept dry, inside and out, no difficulty will arise. But I doubt if it is as well understood that the strains that may be produced within a pipe by the presence of water may be enormously severe. I have had opportunities on various occasions to ascertain how severe are these strains, and how serious, and sometimes how fatal, are the results of settlement of water in a steam-pipe, that may act by condensation of steam causing "water hammer," or may be precipitated in such form that it may be carried over as a "slug," to act like a hammer wherever it may finally strike.

One of my earliest experiences of this sort occurred in this way: we built a mill-engine which was set in the middle of a large engine-room. Steam was carried from the boiler-room adjacent, down the opposite wall and under the floor, a distance of several feet, then up to the steam-chest of the engine. In the *U* thus formed was placed a cock, to be opened for draining it, by the engineer, whenever the engine was stopped, and to be closed when the engine was moving. It happened that one morning the engineer was not in the engine-room at seven o'clock, and his son, who was his assistant, came in, and not finding his father there, at once stepped to the throttle-valve, which was set in the pipe lying against the wall, at the point where the steam entered the *U* on the way to the engine. The instant he opened the valve there was a crash; the cast-iron steam-pipe was broken below the floor. He went below, after shutting off the steam again, and found his father's body; he had been killed by the exploding steam-pipe. He had gone down to set up a joint which probably had been loosened by this very action. This fact illustrates either the force which water may exert when forced through a pipe by the impelling power of the steam, or the forces that may be set in action by the sudden contraction of a moving mass of steam when com-

ing in contact with a mass of cold water. Either action would have been sufficient for the result which I have described. Which actually caused the disaster was never known.

Another instance, in which an accident also occurred, illustrates these tremendous stresses in another way. We had, in the gunboats built for the navy in the early days of the war, a line of steam-pipe that, in some cases, led from the boiler, with a slight downward grade, to a throttle-valve placed on top of the steam-chest. Where not effectively drained, the water would collect in them. On one occasion the chief engineer opened the throttle-valve without seeing that the steam-pipe was clear of water. A mass of this water was carried over into the engine, and broke out the dividing web between the steam and the exhaust ports under the steam-chest. Either of these effects would have required a static pressure of hundreds, and possibly a thousand, pounds to the square inch.

I was called upon to examine the wreck of a steamboat engine on the North River some years ago. I found a steam-pipe, about 24 inches in diameter, leading from the steam dome of one of the common North River flue-boilers downward toward the engine, and turning up under the bed-plate, to the steam-chest. In this bend was placed a cock for draining off water; and, as in the case of which I spoke a moment ago, if this cock were not used efficiently at all times, water might collect. A dispute had arisen between the proprietors and the builders of the engine as to who was responsible; the proprietors saying that the crash, which wrecked the engine completely, was due to some defect, probably the crank-pin. The builders claimed that it was due to the fact that the steam-pipe had not been kept free from water. In examining the steam-pipe I found, in the bend at the foot of the long line of pipe, *tide-marks* on a layer of rust and incrustation, accumulated on the interior of the pipe, which showed conclusively that there had been times when the water had accumulated in this bend. Accumulation could go on in the pipe until, finally, the passage left for the steam became so narrow that the engine could not be supplied through so small an area. The result then would be the production of a pressure on the mass of water which would sweep it up. Seal the orifice, and the steam would pick up the whole volume of water, sweeping over with it into the clearance spaces of the engine. When the piston came back on it, as a matter of course, the engine

went to pieces. I have known in my experience with vertical, beam engines, and with marine engines, several to be completely wrecked by this kind of accident. I presume there are members present who may be more familiar with accidents due to the practice of running high-speed engines from lines of pipe that are not carefully provided with separators or automatic drainage.

It would be interesting to learn how large these stresses are. I doubt if any one has ever found a way of exactly measuring them. But the fact that such accidents occur—and many have occurred, unquestionably due simply to the impact produced by the rapid condensation of steam on the surface of a pool of cold water—shows that these stresses may be enormously great. What may happen when a rapidly moving, heavy mass of solid water, in full career, strikes any obstruction, we all know; but the "hammering" of steam in pipes also produces a local strain, probably quite as severe, perhaps even more serious.

This second kind of strain is known to be enormously great; but we do not know exactly how great. Cases reported by me to this Society some years ago prove a high minimum limit.* I had occasion to examine a quantity of pipe taken out of one of the steam-heating systems then in operation in the city, but since extinct. I was informed that this piping was defective; and the president of the company desired that I should examine them, with a view to reporting to the company on these defects, so as to secure from the makers a reduction of their charges, and possibly damages for the results of whatever claimed to be imperfect welding. After examining a large number, I came to the conclusion that they were not originally defective. I found them split through good welds and bad welds, through solid iron even, and the only report which could be made was that they had been injured by water-hammer. I took a quantity of the pipe to the mill in which it was made, and measured the pressures that could be put upon them, split and weakened as they were. In order that we could obtain a fair idea of the actual pressures that the pipes would sustain, I arranged a rubber packing on the inside of each pipe, a strip covering each crack from end to end, drilling a few holes *along the crack*, so that the strength of the pipe should not be

* *Transactions*, Vol. IV., 1882–3, p. 404; Water-Hammer in Steam-Pipes.

affected, and to insure that sealing these joints should not increase the strength of the pipe. The bolts simply held that packing up against the crack on the inside, so as to seal it by the slight pressure of a line of small bolts which were put in simply for the purpose of holding the packing in place. Pipes arranged in this way, and tested in the hydraulic apparatus of the mill, carried all the way from 300 to 800, 900, in some cases to over 1,000 lbs. to the square inch, *injured as they were.* The conclusion was obvious that the water-hammer to which they had been subjected was enormously in excess of these figures, representing the strength of the pipe after the crack had been made These facts are more impressive than any possible examination, without actual measurement of these quantities, and reveal the intensity of the strains that occur, and the risks of danger which arise, from allowing water to stand anywhere in a line of steam-pipe. After water has once collected in a pipe, especially in the steam-pipes leading to engines of large size, I think there is no safe way of removing this danger except by simply shutting the steam off at once, if it is moving in the pipe, or keeping the throttle shut if it is not moving; then let the steam down, and drain the pipe completely before steam is again put on. If an attempt is made to drain even a still pool of water in a pipe under pressure, the water-hammer may become very severe. The disturbance of the pool by the flow of steam causes condensation; condensation causes a rush of steam upon the surface of the water, and presently there may result as serious effects as when steam actually moves through the pipe with the throttle-valve open, and the pool of water is set in motion to cause accident by impact.

Mr. W. F. Durfee.—I would state some experience I had with some boilers at the Midvale Steel Works. I found two boilers over the heating furnaces in the forge; these boilers were ordinary cylindrical boilers, such as I had often used before. I found that in the case of these particular boilers, every Sunday there was a gang of calkers trying to make them tight. That had been the history of the boilers from the start. I went to the boiler-maker, who was one of the best in this world, I think, and he could give me no information about the cause of the trouble. He had made scores of such boilers, and ours was the first that he had ever had any trouble with. The great bulk of the steam at that time was made in the steel foundry from the

waste heat of the melting holes. Into the main steam pipe were connected the two boilers above-named by branch pipes. These boilers were supplied with water independently of the boilers that generated the most of the steam. I found I could not get any light from any boiler-maker as to the cause of the trouble. After giving the subject some consideration, I found out that these boilers were really evaporating very little water; for there was very little "feed" put into those boilers during the day; and it suddenly occurred to me that it was very possible that those boilers were not making any steam at the temperature at which the boilers in the steel foundry were producing steam, and that what those boilers were really doing was practically condensing steam. The more I thought of the matter the more I was satisfied that I was correct, and, therefore, I pitched one of those boilers out of doors one Sunday. The next week we apparently had just as much steam as ever before, and the Sunday following I pitched out the other boiler, and those boilers were never put back. The moral of that is, do not undertake to boil water by applying heat on its top; if you do, you will surely have a leak from its bottom. Gentlemen were speaking of some receivers that collected the water of condensation, that were very tight under cold pressure and became very leaky under steam. The water that accumulated in them from the drip was being heated from the top while the bottom of the receiver was practically cold; hence leaks, and frequent and emphatic remarks relating thereto.

Mr. George Dinkel.—It seems to me that Mr. Pearson altogether condemns wrought-iron pipe. In places each have their advantage; yet I rather favor wrought-iron pipe for long runs. Now, we had occasion some time ago to run a line of pipe about 300 feet in length, and hung about 50 feet above ground. It was 18-inch pipe, connecting two boiler-houses of about 3,200 and 4,400 H.P. respectively, under about 65 lbs. pressure. The pipe was supported on its ends, and at two places in between. If we were to use cast-iron pipe, I wish you would all consider the extra weight which we would have to support (about 40,000 lbs.), also the risks of sand and blow-holes, and the leaks at chaplets.

It was decided to take wrought-iron pipe, the joints being made thus: we took the pipe with its threads cut, as usually received from the pipe mills, and cut the threads of the flanges, so as to screw up tight, and allow the pipe to screw through or

project beyond the faces of the flanges, as at *a* in sketch (Fig. 160).

The pipe in this manner was placed in the lathe, and the pipe with its flanges faced so as to be perfectly parallel with each other.

By thus screwing the pipe through, and then facing the pipe

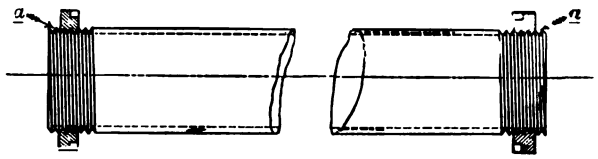


FIG. 160.

and flanges, the pipe itself forms part of the face of the flange (in the above-mentioned case about ½ inch); and in making a joint the ends of the pipe which are bolted together come butt and butt together, as at *b* in the following sketch (Fig. 161).

Now, the leaks on wrought-iron piping are mostly through the threads. But in the above way of putting on the flanges, before steam can leak through the threads it must pass the joint formed by the ends of the pipes themselves. In the case of the 18-inch pipe above mentioned, the pipes, before being put together, were tested to 130 lbs. pressure; and have now been in position since about 1882, and there are no leaky joints on it to-day.

Mr. Pearson has also spoken of the fact that he was hardly able to enter a large steam plant with high-pressure, in New

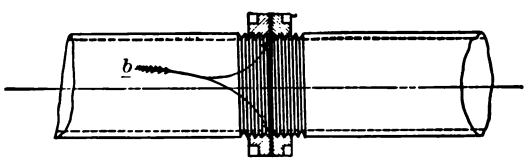


FIG. 161.

York, which was not full of steam, due to leaky joints. If such is the case, it is then time for the engineer to get out.

Leaks seen about large steam plants which run continuously, I think, are more due to the stuffing-boxes of valves than pipe joints. I might say that the plant with which I am connected

has been running almost continuously for the last twenty years. The main steam-valves used are made at the refinery, and are so made that they can be packed when *opened*. The valves being open the greater part of the time, there is no reason for having any leaky stuffing-boxes on the valves whatever.

As to non-conducting covering of pipes: on the pipe mentioned above, it is first given a coat of paper pulp and lime of about 2 inches thick; this is covered with 2 inches of felt, which in turn is covered with 1 inch of straw, and the whole protected by a sheeting of wrought iron. I have repeatedly seen the snow lying on the top of the pipe.

As to the expansion of piping, Mr. Pearson speaks of expansion going by jerks. I have never seen such a thing; but if such has been noticed, I believe the trouble has been that use is made of the common sleeve expansion joints, and that the jerks will be removed a great deal if the man in charge will only look to how he pulls up the glands.

At the sugar refinery we cannot shut off at random to pack expansion joints, and we therefore resort to copper pipe. On the long pipe, above mentioned, the expansion, which amounts to about 2¼ inches, is taken up by copper bends at each end.

The expansion of the main steam-pipe and connections from the Babcock & Wilcox boilers (equal to about 7,600 H.P.) is also taken up by copper bends. The main blow-off pipe, being also connected to boilers by copper bends, allows the boilers to expand freely.

Mr. Taylor has spoken of the quick deterioration of hair felt. If the felt is kept dry, and at the same time kept from being burned, there will be little deterioration. In the above-mentioned case, in which the pipe is first covered with lime and paper pulp, I know cases where the felt is on about sixteen years, and no deterioration has been noticed.

Mr. Frank Richards.—I have little doubt that Mr. Pearson's advocacy of cast-iron piping for high-pressure steam will encounter a general protest, and it is not likely that his success in the application of it will encourage many to follow him in the practice. Mr. Pearson's success with the cast-iron pipe is to be credited entirely to Mr. Pearson, and not at all to the cast iron. He evidently has fallen into the very common error of making two or three changes at once, and then of mistaking the detail to which his success is due. With the change of material

Mr. Pearson also changed the mode of packing the joints, increased the size of the bolts, and, it appears, also applied different expansion joints. Few will be inclined to accept Mr. Pearson's suggestion, that a style of piping that is all right for 120 lbs. pressure is all wrong for 180 lbs.

I would use wrought-iron pipe. Let the flanges be heavy enough, and let them be screwed on until the ends of the pipe come flush with the faces of the flanges, or slightly project (Fig. 162). The ends of the pipe must, of course, be without crack or flaw. After the flanges are screwed on tight, face one end *A* of each length perfectly flat and smooth, and turn a spherical seat at the other end of the pipe *B*, a radial line from the centre of the spherical seat forming an angle of 30° with the

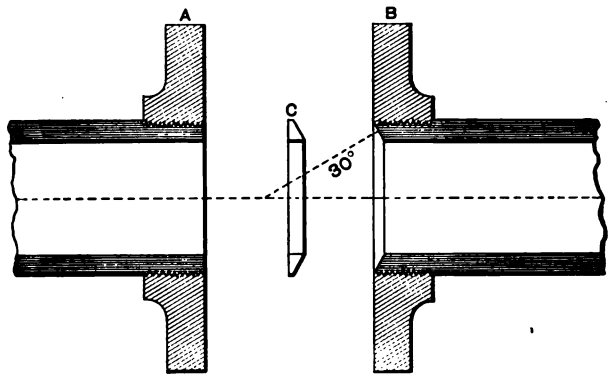


FIG. 162.

axis of the pipe. Then, to make a joint, insert between the two ends of the pipe a turned wrought-iron ring *C*, flat upon one side and spherical upon the other, corresponding to the two ends of the pipe, and tighten up the joint well and evenly all around, and it will be tight, and the pipe need not be absolutely and rigidly straight. The flat face of the collar *C* might be slightly convex, and the spherical face also might bear harder in the middle. If the metallic surfaces are all smooth and true and perfect, no grinding will be required.

Mr. C. W. Baker.—I think we ought to have on record in the *Transactions* a joint which has given great success in England, and which has recently been put on the market in this country. It is very simple, and does away with the use of cast iron except in compression. You know, with the ordinary wrought-iron

joints you have a cast-iron flange screwed on to the wrought-iron pipe, and if your pipe is at all thin you have a screw thread which greatly weakens the pipe. In this joint the wrought-iron pipe is simply expanded at the end, just as you would expand a boiler tube. Between those two expanded ends is placed a cast-iron ferrule. It is used as it comes from the foundry, and does not have to be machined. There are grooves in the ferrule in which is placed rubber or lead packing (Fig. 163). Then, to draw the parts together, a ring-shaped flange which slips right over the pipe is put on each side, made with a bevel to fit the expanded end of the pipe, and the bolts draw these flanges together. This joint has been used for several years in the English collieries, and quite extensively, both for water and for steam, and under water pressures as high as 4,000 lbs. per square inch. I understand that it has shown remarkably good results. A member of the Iron and Steel Institute, when in the country three years ago, was the first to bring it to my attention. He had used it in his own colliery, and had got rid of the trouble before experienced on account of blowing out of gaskets from flanged joints.

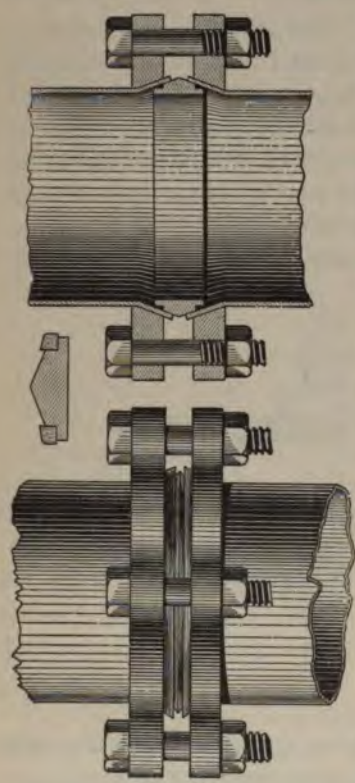


FIG. 163.

*Mr. Wm. A. Pike.**—The very full and interesting discussion on this paper shows that it has fulfilled its object, and with the exception of one or two points spoken of below, there seems to be nothing requiring answer or remark from me. In reply to Mr. Kent's criticism as to my attempt to account for the loss due to vibration of steam in the pipe, I must, of course, admit that the energy is not actually lost, and perhaps reappears in dryer

* Author's Closure, under the Rules.

steam; but it still seems to me that the diagram represents the energy in the form of work used in overcoming the inertia of the steam and resistance of the pipe to the sudden and rapid motion of the steam. I also admit that the friction resistance varies from its maximum to zero at the other end of the pipe, but it seemed to me that the loss per square inch times the area of the pipe into the distance the steam would move through the pipe, if it moved as under the influence of a piston, would represent the lost work.

However, as stated in the paper, this portion was inserted only as an *attempt* to measure the loss, and is of no special consequence, the real object of the paper being to bring out a discussion on piping, which I am glad to see it did.

With reference to the use of cast iron for piping, I am of the opinion that it should be avoided wherever possible, though it must be admitted that at present there is no feasible way, at least of general application, of doing away with cast-iron joints and elbows. This, of course, prevents us from getting the full benefit, as far as safety and strength is concerned, of the use of wrought iron or steel.

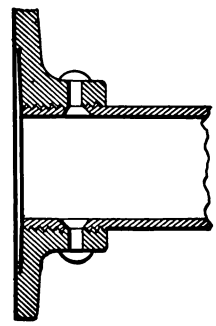


FIG. 165.

Nevertheless, by using cast-iron connections of extra strength, and giving each one a most careful inspection, it is believed that as a whole a much safer system can be planned when using wrought iron than cast. In the sketch above (Fig. 165) is shown a method of making connections which has proved satisfactory, viz., to screw extra strong cast-iron flanges on to the wrought-iron pipe, and also to rivet them together. The connections are made with male and female joints, with good packing. These joints have been found to remain tight after once bringing the joints to a good bearing. I believe that the principal objection to cast iron is the fact that when rupture takes place it is always very suddenly, while wrought iron, in many cases, gives warning by small leaks of steam, thus giving time to shut down and avoid accident. This was particularly brought to my attention in the case of some spiral weld pipe which gave way at the connections in several instances, but, I believe, invariably gave warning beforehand by the hissing of small leaks of steam, gradually growing greater,

until rupture occurred by the slow pulling out of the pipe from the flange connections.

As to expansion, it has been my experience that in most cases there are enough necessary bends, with good lengths of pipe between, to provide for it without other means. In other cases such expansion joints as Mr. Pearson refers to must, of course, be used.

Professor Thurston's remarks as to the great force of a "slug" of water in steam-pipes brings to my mind the fact that after the "explosion" referred to in my paper I tested a piece of the same pipe with blank flanges put in exactly as the others were, and found that it took 800 lbs. of water pressure to start the joint, showing that the water must probably have exerted more than that amount of pressure.

DLXXVIII.*

A METHOD OF MANUFACTURE OF LARGE STEAM PIPES.

BY CHAS. H. MANNING, MANCHESTER, N. H.
(Member of the Society.)

THE rapid increase in size of steam plants and of the pressure of steam used in recent years has of necessity changed the design and material of steam pipe. For stationary engines with pressure of from 60 to 80 lbs. per square inch, cast-iron was the generally accepted pipe, whilst for marine purposes copper still holds its place, though the method of manufacturing has changed with the great increase of pressures; but at best the copper pipe in large sizes is dangerous, and has cost many lives in the last few years.

Eleven years ago it became necessary for me to construct several thousand feet of 20-inch steam pipe, besides considerable lengths of 15 and 12 inch sizes, and the satisfactory results obtained lead me to describe the methods of construction. It was decided to make a mild steel, riveted pipe, with die-forged flanges of the same material, and rivet and calk the flanges. For riveting the pipe an Allen pneumatic riveter was used; and as the extreme reach of the arms of the machine used was about 70 inches, the courses were made of that length, netting between roundabout seams 68 inches. I had observed that the greatest difficulty with riveted pipe was keeping the roundabout seams tight, and this was evidently caused by water accumulating about these seams on the bottom of the pipe when steam died out; then, when steam was readmitted to the pipe, these small accumulations of water caused unequal expansion, as the portions covered by them heated much slower than the unprotected surfaces, and in this way leaks were started in the roundabout seams at the bottom. To avoid this, each course was

* Presented at the New York Meeting (December, 1893) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

made the frustum of a cone, thus bringing the laps all in one direction, and by placing the large end of the cone up-hill and giving the pipe sufficient grade, it was easy to keep it free of water at all times, and this has been entirely successful. (See Fig. 166.)

To insure rigidity and tightness, $\frac{1}{4}$-inch thickness, with double-riveted longitudinal seams and single-riveted roundabouts, was settled on, which of course gives a very large margin of strength; but if building for 160 lbs. pressure or above, I should double-rivet the roundabouts also. The plates are received from the mill sheared to the required taper. They are then scarfed at corners, punched to template, planed on outside edges for calking, and then are rolled to circle on rolls having the outer end housing hinged so that it can be let down and the pipe slipped off the roll. For convenience in handling, the lengths are limited to nine courses, making the total length about 51 feet 2 inches, and in these lengths it has been shipped as far as Nashville, Tenn., without mishap. The flanges, except where coming on to valves or other castings, are of $\frac{3}{8}$-inch stock, and for above exceptions $\frac{1}{2}$-inch stock. They are formed on a cast-iron die with iron plug driven through them with drop-hammer and are then trimmed on both edges in the lathe. To avoid making flanges of two different-sized holes to fit the two ends of the pipe, the last course of the pipe at the down-grade end is a parallel course—that is, it is not coned, and is of the size of the large end of the cone. The longitudinal seams are placed quartering 45° from top of pipe, with the laps pointing up so as to be readily accessible for calking. (See Fig. 167.) When in place the flanges are riveted together, and it is preferable to have all valves made with long necks and rivet them in place, so as to avoid all packed joints. Before leaving the shop, headers are bolted to each piece, and it is then calked under a steam pressure of 100 lbs.

The quarter turns for 20-inch pipe are made of 3-foot radius of axis, and are formed from two sheets on a cast-iron former, and have, when complete, one row of rivets along the back and the other along the throat. (See Fig. 166.) Owing to their rigid form, single-riveting is found to secure tightness; but to allow for drawing of metal in forming, $\frac{5}{16}$-inch stock is used instead of $\frac{1}{4}$-inch. The tees are made of three sheets, which brings all the riveting on the sides, and are made over formers similarly to

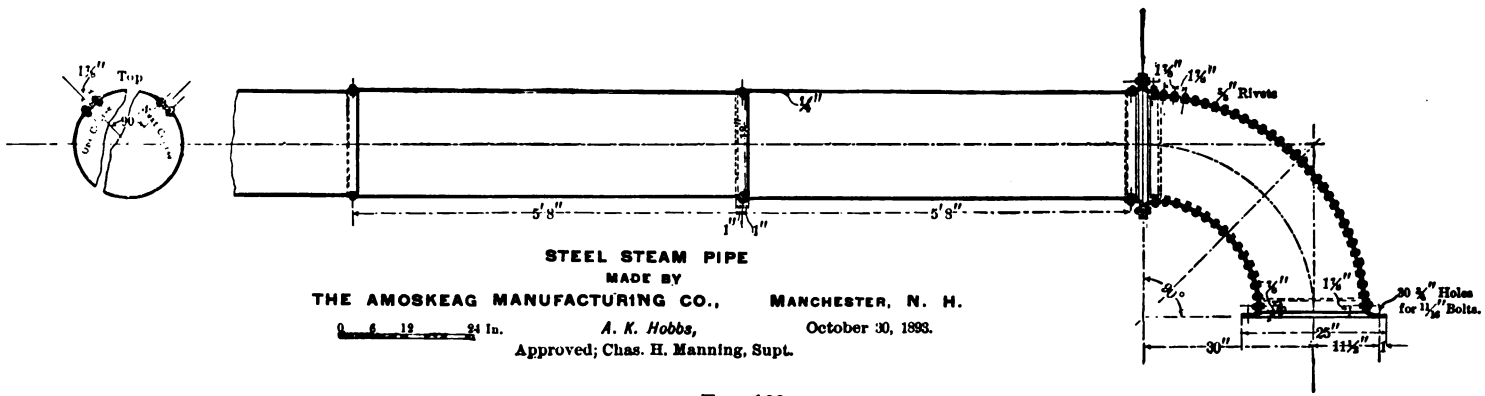

1⅞"
Top
90°
1/4"
5'8"
1"
1"
5'8"
1⅞"
1⅞"
5/8" Rivets
45°
7/8"
1⅞"
30 3/4" Holes for 11/16" Bolts.
25"
30"
11⅛"
STEEL STEAM PIPE
MADE BY
THE AMOSKEAG MANUFACTURING CO., MANCHESTER, N. H.
0 6 12 24 In.
A. K. Hobbs,
October 30, 1893.
Approved; Chas. H. Manning, Supt.

FIG. 166.

the quarter turns. In the flat spandrels sufficient socket stay bolts are used to insure strength. For branches up to 6 inches diameter a threaded composition saddle is riveted to the pipe, but for larger sizes a short piece of pipe of the desired size is made by welding up from sheet metal, and then this is flanged in a cast-iron former to fit the pipe, and riveted on. At the other end a thread may be chased, on which a cast flange may be screwed, or a wrought flange may be riveted on, which latter I prefer.

This pipe may be hung in chains, but where practicable I prefer to support it from underneath. With the 20-inch size it should be supported about every 17 feet, and it is my practice to support the 51-foot lengths in three places, at the middle having a sliding foot consisting of a foot clamped firmly to the pipe and sliding in ways secured on the foundation, and toward the ends by rockers secured to the pipe and rocking on a cast-iron plate. Where expansion is to be provided for, I use a carefully constructed slip-joint, the female portion being provided with legs which slide in guides similar to those of the sliding support of the pipe. These slip-joints are packed with braided asbestos packing, and I find no trouble in keeping them perfectly tight with about one new turn of packing per annum. All large pipes are liable to water hammer, and a hammer which would completely wreck a cast-iron pipe or split a lap-welded pipe from end to end will have little or no effect on the riveted pipe, except to strain the joints of the longitudinal seams; but I have never known a riveted pipe to give out.

This method of making the laps all in one direction is not new, though I supposed it to be so for several years after I began to use it; my belief was changed by coming across a piece of similarly constructed 12-inch pipe in a scrap heap, the piece having the appearance of being twenty or thirty years old, but of course the courses were much shorter than I use. Some makers of large riveted pipe use very unsightly cast-iron flanges, which they seldom succeed in keeping tight for any length of time, and the cost is fully as much as with the wrought flanges. The accompanying illustration is of the exhaust piping of a pair of 2,000 H. P. Corliss engines, piped to use one or both cylinders condensing or against back pressure, as desired.

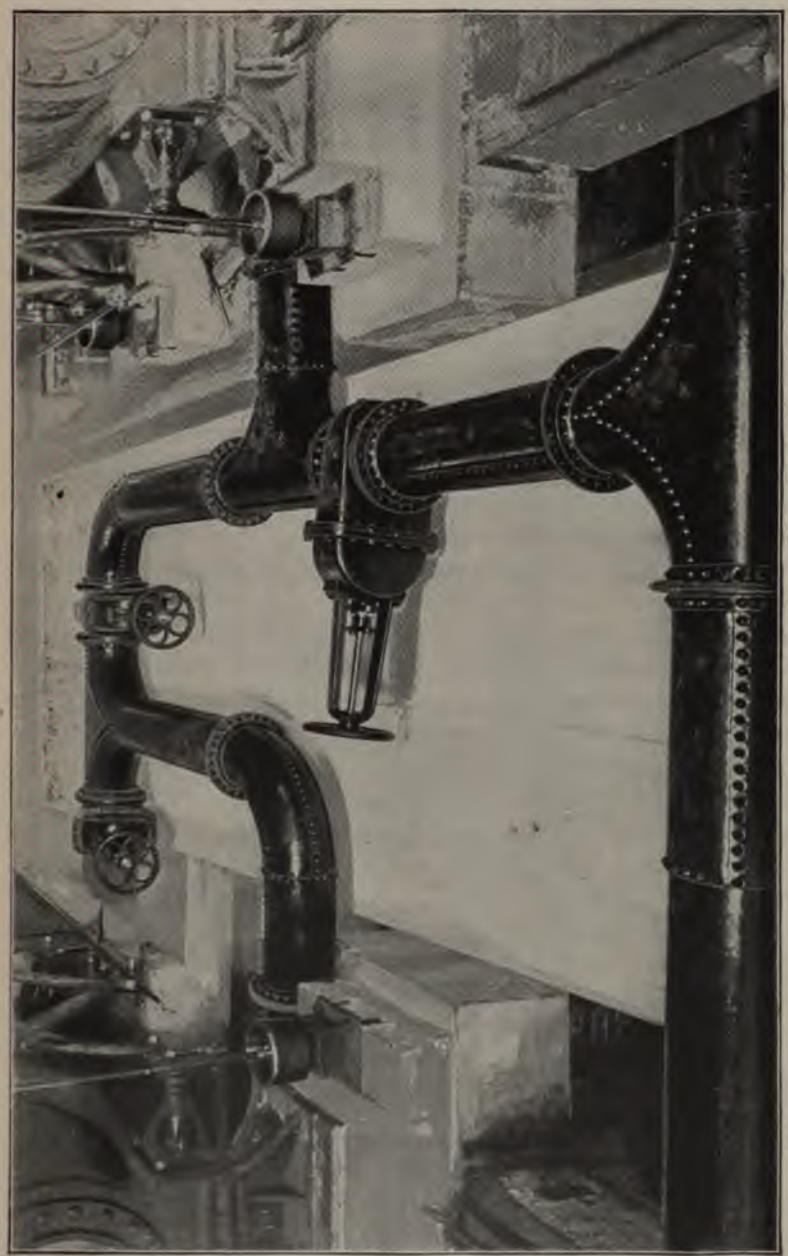


FIG. 167.

DISCUSSION.

Mr. Chas. H. Manning.—The lateness of the hour, and the full discussion of the preceding paper by Mr. Pike on "Steam Piping" * in which the topics of this paper also received considerable comment, render unnecessary any extended closure. I was well aware when I wrote the paper that the limit of 100 lbs. of steam pressure was fast going into ancient history. The scheme was laid out eleven years ago, and has been thoroughly successful, and the pipe is tight, without any trouble from leaking. I think if the member who spoke in debate on "Steam Piping" had continued his investigation a little further at the time he was making it, I could have shown him a pipe which would have certainly made him think more before he went back to cast iron. I should be very loath to take the back track myself. Even with 160 or 250 lbs. pressure, I would much rather do a little calking once in a while than take my chances with water hammer, with 250 lbs. of steam, on any cast-iron pipe ever made. This gentleman also stated, as bearing on this subject, that he had a steam drum or separator made of $\frac{3}{4}$-inch steel so as to be sure to get it tight. I think there is where he made a mistake. The $\frac{3}{8}$-inch steel was amply strong for the diameter, as I understand it, for any pressure ever to come on it, and it is very much easier to make thoroughly good boiler work with $\frac{3}{8}$-inch steel than it is with $\frac{3}{4}$-inch steel; but when the pressures necessitate using $\frac{3}{4}$-inch steel the methods and means can be found to make it perfectly tight under that pressure without trouble. That there have been a great many bad wrought-iron riveted pipes, there can be no question. They have given immense trouble, and in some large plants have been taken out very recently; but it is not the fault of the material—it is simply the fault of construction. They were made on bad principles, and erected on worse. There is no trouble in making thoroughly tight steel riveted pipe to stand any pressure used in steam engineering to-day.

* *Trans.* A. S. M. E., Vol. XV., page 536, No. 577.

DLXXIX.

TOPICAL DISCUSSIONS AND INTERCHANGE OF DATA.

XXVIIIth MEETING.

No. 579—99.

Given, a factory driven by steam power, where the power used varies greatly, even as from 1 to 4, as in a rolling mill.

It is desired to ascertain the average power used here for a whole day.

Which is the better of the following ways?

First, To take single indicator cards at fixed intervals of time by the clock, and get the average of these; or,

Second, To leave the paper on the indicator and allow it to run for a minute or two; then follow it quickly by another, and so on, continuously; then judge by the color of the band of pencil marks where the average line would run, and calculate this power.

Is there any better way than one of these?

Mr. Jos. C. Platt.—I would like to say, as an explanation of why I sent that question in, that I had occasion some time ago to ascertain the power used in a certain factory. There was a test made, and some were in favor of taking a continuous card or a series of continuous cards. Others were in favor of taking cards by the clock every minute, or two minutes and a half, or five minutes. There was a marked difference of opinion with reference to the question, and several gentlemen of very high standing said that they had never heard of taking continuous cards. Others said it was a common thing. So, seeing this list of papers having so much about indicators, it occurred to me that this would be a favorable opportunity to put the question before the Society and get the opinions of other members. My own preference was in favor of letting the indicator run for a minute or more, and then seeing where the heaviest mark was, because where between two revolutions of the engine the card jumped from friction card to maximum card, a person who happened to strike one would certainly not get an average; neither would the other. I would like to know what the gentlemen here think of the subject; and if any one thinks there is any better way, I would like to know what it is.

Mr. Wm. Kent.—I think the plan of taking a continuous card is the best under the circumstances. I had occasion once to make tests of a throttling engine, under a variable load and a poor governor, and I kept the pencil on the indicator card perhaps ten or fifteen strokes, and then took the average of the lines. In that case the lines made by the pencil usually made a band, say a quarter of an inch thick; that is to say, the lines were pretty nearly uniformly distant. But I can see that with loads varying irregularly, the heaviest, or most numerous, pencil lines might be in one part more than another, and then the heaviest shade of pencil lines would determine where the average was. I think this method is better than the method of taking by clock at definite intervals; for it might happen that the time of taking the cards would be synchronous with the times of greatest load or least load, and the average of the cards might be far from representing the true average horse-power.

Mr. Geo. I. Rockwood.—I think the subject under discussion is very interesting.

Where the load is pretty nearly constant, it is my own practice to use the card as suggested by Mr. Kent. In the tests of the Oakdale engine, reported in connection with Professor Thurston's paper, I held the indicator pencil on the diagram for a period of half a minute. This resulted in cards having single lines for back-pressure, compression, and steam admission, but in expansion lines which were from $\frac{1}{32}$ to $\frac{1}{16}$ inch thick. Thus the horse-power, as measured, must have been more nearly correct than if we had taken but a single diagram.

Where the variations in load are excessive, as in electric railroad or rolling-mill work, there is but one correct way of indicating the power exerted by the engine, and that is to obtain a diagram for every stroke. I saw recently a long roll of paper, which proved to be a continuous indicator card, showing distinctly and separately a diagram for each stroke of the piston. I thought that the method of obtaining this continuous diagram, or series of diagrams, was by no means perfect, although the effort to obtain such a series was certainly in the right direction. A machine which would satisfactorily accomplish this result would fill a want.

Prof. J. B. Webb.—I cannot see that taking the heaviest mark would be correct. Suppose the indicator to have made a rectangular card with a gradually decreasing pressure, so that the

pencil went over the same track at the bottom and ends of the card, while at the top it drew a series of horizontal lines, resulting in a band of lines, say, a quarter of an inch wide. If the lines are equidistant you have a simple band, and would be justified in taking the centre of it for the average card. Suppose, again, that you commenced taking the cards just before the pressure commenced to fall, or that you continued taking them after it had stopped falling, so that the pencil ran over the same line three or four times at the top of the card, or bottom, as the case may be. This will be enough to make a dark line appear at the top or bottom of the band, whereas the correct average would not differ much from the centre line of the band. In fact, if the number of revolutions in the case of the simple band, whose centre line is its correct average, be thirty, and if, by adding six revolutions before the pressure commences to fall, the heavy line at the top of the card is made, then the correct average will be a line distant from the centre line by only one-twentieth of the width of the band, so that the heavy line at the top of the band would in this case be very much in error. So it is evident to my mind that you might make a great error in taking the darkest place in a band of that sort.

Mr. W. F. Durfee.—I do not know that there is any instrument manufactured and on sale which will make indications of this kind. But a number of years ago the French engineer Morin designed and had built an instrument called a "compteur" for registering continuously the traction of loaded carriages upon common roads. The general principles adopted in this instrument were first proposed by M. Poncelet. Following out the general ideas of MM. Poncelet and Morin, Professor Moseley designed an instrument which he called a "continuous indicator," which made a record for every revolution of the engine. Any one interested in looking the subject up, will find a large steel plate engraving of it in Vol. I. of the *Practical Mechanic and Engineer's Magazine* published in Glasgow in 1842. The instrument seems to have been very carefully studied out, but for some reason or other it has never come into general use.

Mr. C. S. Dutton.—I have had occasion to take a good many indicator cards in rolling-mill practice—I speak particularly of that—and I think I am perfectly safe in saying that the method of taking cards at regular intervals in rolling-mill work is ab-

solutely unreliable. There is nothing approaching an average of the power developed to be ascertained in that way, when you take into consideration the character of the work. Perhaps an ordinary case would be an engine having a maximum power of 500 H. P., in which the rolls running light would indicate about 50 H. P. That means the heaviest pass through the rolls requires 500 H. P. It depends altogether on the character of the order and the way in which the rolls are handled as to the average of the power. The great majority of the time the engine is running very light, and the cards I have taken were usually not to ascertain the average power required, but to determine the maximum, which is what is required in designing. For rolling-mill work you have got to have an engine large enough to do the maximum work. What is the most economical load for it has nothing to do with the case. Holding a pencil on continuously for some time you can make a better guess as to the average work than you can by the other method. Probably by holding it continuously and measuring every card you can get a reasonably accurate average.

Mr. F. H. Ball.—It seems to me this method might be very misleading in a good many cases. Suppose on a card taken in this way the lines were all blended together, so that the average diagram would be represented by the centre of the broad line, and suppose that during most of the time the load had been either the maximum or the minimum—don't you see it would be very easy to get a result entirely at variance with the truth? It seems to me we would be much surer of getting accurate results by taking individual cards, and if the load is very intermittent, taking them continuously, following one right after another. If we take the maximum load and the minimum load, and average the two, it might be very far from the truth.

Mr. J. C. Platt.—My use of the word "average" was perhaps misleading. I did not mean what you might call the average between the highest and the lowest when I used that word, but the power which was used most of the time. I can make it perhaps a little plainer by telling you what the factory was. There was a double 500 H.P. Corliss engine, and this engine drove a large number of machines which, perhaps, took one-tenth of a horse-power, and a machine which took 100 or 150 H.P. This machine which took 150 H P. was very much like the roughing-down train in a rolling-mill. It called for very heavy

power for a few seconds, and then nothing except friction. So the question came up as to what was the power used in that factory, as to how much it would take to run that factory as a regular thing. I gave an opinion that it would require at times 700 H.P. to run that factory up to full capacity, and at other times they would not use over 150. With that explanation you will understand just what I had in mind. One card taken showed about 52 H.P., while others were about as near square as can be taken from a Corliss engine.

The following cards are reproductions of some of the continuous cards taken:

FIG. 168.

FIG. 169.

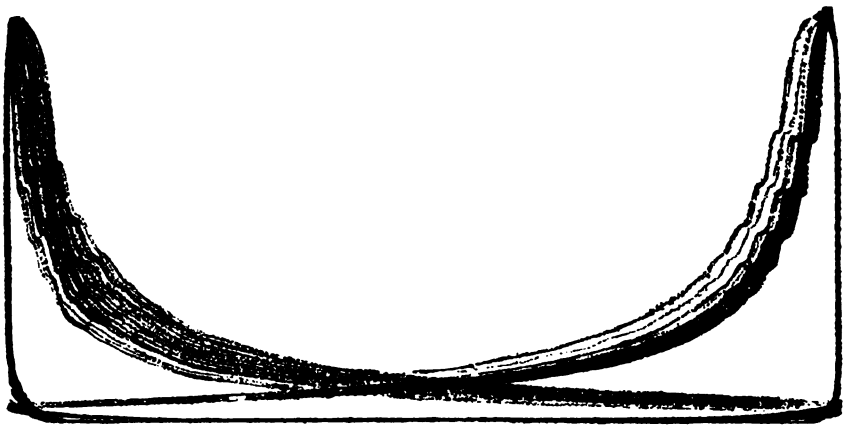


FIG. 170.

Mr. F. R. Low.—Some five years ago I had made by the Ashcroft Manufacturing Company an indicator in which the pencil was replaced by a planimeter wheel the angular position of which with reference to the axis of the drum was varied by the movement of the indicator piston. Instead of taking a diagram the indicator revolved the wheel an amount proportional to the area of the diagram which would have been taken had the pencil been used. By a train of recording dials the revolutions of the wheel were counted, and it could be run for a minute or an hour if need be, and give you at the end of that time the same result that you would get by taking a diagram for every stroke, measuring each and adding their areas together. It did very well at slow speeds, but high-speed engines ran away with it.

Mr. Daniel Ashworth.—I recollect seeing the indicator referred to by the gentleman. It was known as the Darke indicator.

I would say that the plan of taking a series of diagrams, not particularly to obtain the average, but to obtain the power developed by the different passes, is one we are continually following up. It is my preference to take a single card; whilst we have no trouble whatever in obtaining the others with accuracy, it gives us an opportunity of ascertaining in detail the special pass without the confusion of papers afterwards. Our blooming mills, rolling mills, and rod mills, and especially our rod mills —and you had better put the blooming mill also upon the same plane—require considerably closer attention, promptness of

signalling and rapid working; but we have no trouble whatever in getting the power developed accurately and very distinctly of the various passes, even by these high-speed operations, and they are an interesting study.

Mr. George Richmond.—In this connection of continuous indicators I am reminded that I remember distinctly having seen some twenty years ago, in an instrument maker's in Manchester, England, of I think the name of Storey, a continuous indicator which I think was an integrator at the same time. I do not know that I am correct about the name, but I have a very vivid recollection of seeing that instrument and watching it from time to time. The store was near the cathedral.

Mr. Ashworth.—I would say that this continuous instrument to which I refer was made by the Elliott Brothers of London.

Prof. R. C. Carpenter.—The gentleman no doubt refers to the Ashton & Storey continuous power meter. This instrument was made some years ago, in England, but judging from the crude methods employed in the manufacture, I do not believe it ever was a commercial success. Dr. Charles E. Emery purchased one of these instruments, and about two years ago presented it to Sibley College. This particular instrument has about the form and size of an old-fashioned eight-day clock. It consists of cylinder and piston, a reducing motion, integrating mechanism, and a dial on which is recorded the work of the engine. Steam pressure acts on both sides of the piston; its motion is resisted by a spring, so that its travel is proportional to the pressure acting in the engine cylinder. This acts on the integrating apparatus, which is also moved by the reducing mechanism, in such a manner that the resultant motion is proportional to the work. The dial is graduated to show foot-pounds of work per circular inch of piston.

A very much improved form of this instrument was exhibited at the Inventions Exhibition, South Kensington, in 1885, and received a high reward. This instrument was known as Ashton's New Steam Power Meter and Continuous Indicator. It was manufactured by Ernest Scott and Mountain, Limited, Newcastle-on-Tyne. From a private letter to Dr. Thurston I understand that the instrument is no longer manufactured. Because of the importance of such an instrument for engine testing, in connection with street railroad work, and of the manifest interest in the

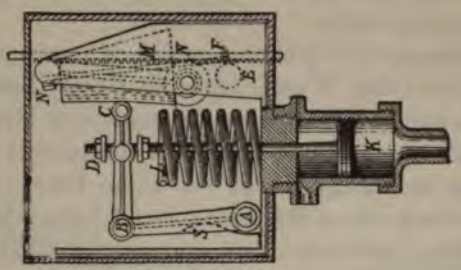


Fig. 173.

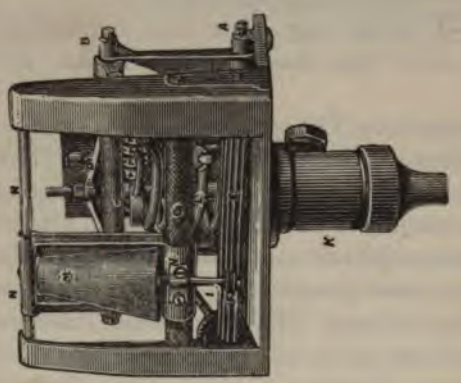


Fig. 172.

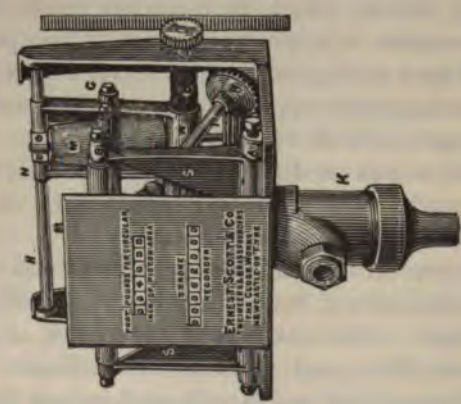


Fig. 171.

Like letters represent corresponding parts in all three figures.

subject at this time, I have submitted the following description and demonstration of principles of operation from the circular issued from the former factory.

"The indicator, as illustrated above, is composed of a small double-acting cylinder *K*, each end of which is connected with the corresponding end of the steam engine cylinder; the motion of the piston and piston-rod of this small cylinder is controlled by the action of a spring *L*, which limits their range of movement from the middle of the cylinder either way to a distance proportional to the pressure on the piston.

"To this piston-rod is connected a cylindrical roller *C* by the intervention of levers and crosshead *A B D*, so that the axis of the roller can move in a plane at right angles to the axis of the piston-rod and keep itself parallel to one and the same straight line in that plane, and also to move with a velocity bearing a fixed proportion to that of the piston-rod. The convex surface of this roller is covered with teeth. A hyperbolic conoidal drum *M* is carried in a frame with bearings at *N* and *N*, which frame can slide along the guides *G* and *H*, which are parallel with each other, and also with the axis of the roller *C* in all its positions. The convex surface of the conoidal drum *M* is also covered with teeth similar to, and of the same pitch as, those on the roller *C*.

"The plane through the axis of the conoid *M* and through its point of contact with the roller *C* being at right angles with the axis of that roller, and the pitch of the teeth being the same in each, while the continued pressure of the spring *S* transmitted on to the roller *C*, will by means of these teeth give continuous interlocking connection between the roller and conoidal drum, and if the frame *N* and *N* carrying the conoidal drum *M* be now slided to and fro along the guides *G* and *H*, rotary motion will be imparted to the drum in a direction corresponding to the direction of that sliding motion; and with a velocity dependent upon the position of the roller *C* on the tapering surface of the conoid. The frame *N N* carrying the conoidal drum is to be connected with the crosshead or any other proper and convenient reciprocating part of the steam engine, so that it may carry the conoidal drum in one direction along the face of and in contact with the roller during the advancing stroke of the engine piston, and in the other direction and in contact with the roller during the return of the above piston, and in such a

manner that its speed of motion at any instant has a constant ratio to the speed of the piston at that instant.

"When therefore the engine is not in action the indicator piston will be at about the middle of its cylinder, and the spring L which controls its motion will be in its normal state of rest, with the roller C in contact with about the middle point of the conoidal drum.

"On steam being admitted to one side of the engine piston, and as a consequence to the corresponding side of the indicator piston, the last-named piston is moved in its cylinder until the force tending to move it is balanced by the resistance of the spring L, the aforesaid roller C being at the same time moved along the surface of the conoid to a distance from its middle proportionate to the pressure of the steam on the piston; and as the said conoid is moved along with its bearings by the frame $N\ N$ being slided by the engine crosshead along the face of the cylindrical roller C and in toothed contact with it, a rotary motion will thereby be imparted to the conoidal drum, the velocity of rotation being dependent on the speed of the crosshead of the engine, and the position of the roller C on the conoidal drum, or in other words on the speed of the engine piston and the pressure of the steam on it.

"And in order that the differences of angular motion imparted to the conoidal drum may be proportionate to the corresponding differences of pressure on the piston during a certain and constant amount of its movement along the face of and in contact with the cylindrical roller, the diameter of each circular section of the conoidal drum at right angles to its axis must be that of a solid generated by the revolution of a hyperbola about one of its asymptotes.

"Then during the forward stroke of the engine piston the conoidal drum is moved, say, forward along the roller and in toothed contact with it, while at the same time the roller is held by the pressure of the steam, say, above the middle of the conoid, by which means the conoid is driven rapidly round in one direction during that forward stroke of the engine piston; while during the return stroke of the engine piston the conoidal drum is moved backwards along the roller and in toothed contact with it, and at the same time the roller is held by the pressure of the steam below the middle of the conoid, by which means the conoid is driven slowly round in the opposite direction during that

backward or return stroke of the engine piston, the difference of the angular motions of the conoidal drum between the forward and backward strokes of the engine piston being strictly proportionate to the space traversed by that piston multiplied by the pressure on the same, which difference is, therefore, strictly proportionate to the power developed by the engine during that double stroke, the same being true for every other stroke.

"A wormwheel F, connected with the conoidal drum, gives rotation to the long pinion E, and thence through I motion is given to the indices of the recorder by the wheel R. And because the teeth of the long pinion E have a rifled direction given to them equal to one pitch of tooth in the length of the circumference of the middle section of the conoid, the result of the above arrangement is that when the roller C is on the middle section of the conoid no motion is communicated to the long pinion by any sliding movement of the frame carrying the conoidal drum, while when the roller is above the middle of the drum and the drum is at the same time carried forward along it, thereby rotating it rapidly in a forward direction, it is at the same time driving the long pinion E round in a forward direction and also the recording indices in the same direction. Again, while the roller is below the middle section of the drum, and the drum is at the same time carried backward along it, and thereby rotated slowly in a backward direction, it will at that same time also drive the long pinion round in a forward direction as before, as also the recording indices, the record given by the indices during the advancing and returning strokes of the piston being proportional to the work done by it, the middle section of the conoid being in effect the zero point of the recording arrangements as well as the zero point of pressure.

THE FORMULÆ UPON WHICH THE CONSTRUCTION OF THE INDICATOR IS BASED.

"Having thus fully described the various parts of the meter, it may be advisable to give the mathematical calculations which prove the truth and accuracy of the principles upon which the instrument is based. They are as follows:

"The equation to the hyperbolic conoid referred to its asymtotes is $(a + x)\,y = b$; in which a and b are constant, and y dependent on the value of x.

"Then, as above, $y = \frac{b}{a + x}$ = radius of conoid at distance x from its base.

"Let us assume $a = 4$ inches, and $b = 8$ inches:

Then, when $x = 0$, $y = \frac{8}{4} = 2$, or radius of conoid at base is 2 inches.
And when $x = 1$, $y = \frac{8}{5} = 1\frac{3}{5}$ = radius 1 inch above base.
And when $x = 2$, $y = \frac{8}{6} = 1\frac{1}{3}$ = " 2 inches " "
And when $x = 3$, $y = \frac{7}{9} = 1\frac{1}{7}$ = " 3 " " "
And when $x = 4$, $y = \frac{8}{8} = 1$ = " 4 " " "

"Let the length of the stroke of driving roller C across the face of the conoid = S inches.

"Then the number of revolutions, or parts of a revolution, given to the conoid by the roller C during one single stroke of the engine, will be $N = \frac{S}{2\pi y}$, where $\pi = 3.1416$.

"Suppose the roller to be in contact with the conoid in a plane one inch above its base during the advancing stroke of the engine, and in contact with its base during the return stroke:

Then $N = \frac{S}{2\pi \times 1\frac{3}{5}}$ = angular rotation during advancing stroke.

And $n = \frac{S}{2\pi \times 2}$ = angular rotation during returning stroke.

And, therefore,

$$N - n = \frac{S}{2\pi}\left(\frac{1}{1\frac{3}{5}} - \frac{1}{2}\right) = \frac{S}{2\pi}\left(\frac{5}{8} - \frac{4}{8}\right) = \frac{S}{2\pi} \times \frac{1}{8} =$$

rotation for double stroke.

"Suppose roller now to be 2 inches above base of conoid during advancing stroke, and at base during return stroke:

Then $N - n = \frac{S}{2\pi}\left(\frac{1}{1\frac{1}{3}} - \frac{1}{2}\right) = \frac{S}{2\pi}\left(\frac{3}{4} - \frac{2}{4}\right) = \frac{S}{2\pi} \times \frac{2}{8} =$

rotation for double stroke.

Suppose roller now to be 3 inches above base of conoid during advancing stroke, and at base during the return one:

Then $N - n = \frac{S}{2\pi}\left(\frac{1}{1\frac{1}{7}} - \frac{1}{2}\right) = \frac{S}{2\pi}\left(\frac{7}{8} - \frac{4}{8}\right) = \frac{S}{2\pi} \times \frac{3}{8} =$

rotation for double stroke.

Suppose roller now to be 4 inches above base of conoidal during advancing stroke, and at base during the returning one:

Then $N - n = \frac{S}{2\pi}\left(\frac{1}{1} - \frac{1}{2}\right) = \frac{S}{2\pi}\left(\frac{8}{8} - \frac{4}{8}\right) = \frac{S}{2\pi} \times \frac{4}{8} =$ rotation for double stroke.

So we see that the amount of rotation given to the conoid is strictly proportionate to the traverse of the driving roller along the axis of the conoid, and the amount of rotation of the conoid is also proportionate to the traverse of the conoid along the axis of the roller.

"We will suppose our spring L to distend or contract one inch with the application of pressure either way to the amount of 10 lbs., and which we will call a 10-lb. spring. But we may require other and much stronger springs to measure the pressure applied to the various engine pistons requiring to be tested; so that if a 20-lb. spring be applied in the machine, and the same record is given on the indices in the same time as before, we shall know that now double the amount of work is done in the same time, or with a 30-lb. spring triple the amount, or with a 40-lb. spring quadruple the amount, and so on. We shall therefore call the 10-lb. spring No. 1, the 20-lb. spring No. 2, 30-lb. spring No. 3, 40-lb. spring No. 4, and so on, to, say, 150-lb. spring No. 15.

"We see that to obtain the amount of power developed the record on the dial requires to be multiplied by the spring's number which is in the machine at the time. Again, we will suppose that the stroke of the engine piston be 5 feet, while the traverse of the frame carrying the conoidal drum be 5 inches; had the traverse of that frame been greater or less than the supposed 5 inches, the record given on the indices would have been proportionately greater or less. So that taking

d = diameter of engine cylinder in inches;
l = length of engine stroke in feet;
t = length of traverse of drum frame in inches;
n = spring number;
R=reading of dial; and
r = any previous reading, each unit on the dial being taken to represent 1,000 foot-pounds;

m=minutes during interval between readings; and

33,000 foot-pounds per minute to represent one horse-power:

$(R - r) \times 1000 \times n \times \frac{l}{t}$ = foot-pounds of effect per circular inch of piston's area, between readings R and r; and

$(R - r) \times 1000 \times n \times \frac{l}{t} \times d^2$ = total foot-pounds during interval between readings R and r. And, therefore,

$$\frac{(R - r) \times 1000 \times n \times \frac{l}{t} \times d^2}{m \times 33{,}000} = \text{effect in horse-power}\cdot$$

or,

$$\frac{(R - r) \times n \times l \times d^2}{m \times 33 \times t} = \text{effect in horse-power.}$$

"And because n, l, d^2, 33, t, are each constant in one and the same engine, we may put

$$\frac{n \times l \times d_2}{33 \times t} = C,$$

and

$$\frac{(R - r)\ C}{m} = \text{horse-power.}$$

"So that in any case we multiply the result on the dial by the constant C, and divide the product by the time in minutes for the power in horse-power effect."

No. 579—100.

Does copper undergo a change from prolonged service which renders it unsafe after a while to carry high-pressure steam in pipe or expansion joints, etc.?

Mr. Gus. C. Henning.—In a series of investigations made in Austria in 1890 on the subject of the deterioration of copper and brass tubes in boilers, the committee reports that in the course of time the material does change under the action of gases or heat, or both combined, but the exact amount of the change is not given. I think the report will appear in print in a short while, and then it can be seen what there is in it.

Mr. W. F. Durfee.—I think it would be very important, in connection with investigations relative to this matter, to have an analysis of the metals of which the pipes were made. It is not

by any means certain that everything which is sold and used in the manufacture of steam pipes is copper. Copper is liable to contain a number of impurities which may affect in a very great degree its ability to endure pressure under heat for any great length of time, and no examination or test would be complete, in my judgment, unless a very careful analysis was made of the metal tested as to its chemical composition, so that we may know whether we have got in use copper, or copper and something else.

Mr. W. E. Crane.—Copper becomes hard and brittle by bending, or from any blows of any kind.

No. 579—101.

How far can hydraulic (or other) presses be advantageously substituted for the drop press, in the manufacture of small stamped metal articles, either by hot or cold treatment; especial regard being had to the making of plated wares, and to the cost of maintaining the forming dies?

Mr. William Kent.—Mr. President, I have made a rough calculation to show the difficulty of adapting a hydraulic press as a substitute for the drop press. Suppose we have to emboss a surface in sheet metal through a maximum distance of one-eighth of an inch, and we have a 200 lb. drop falling 50 inches, which accomplishes the work. Assuming no give to the anvil, and that all the work goes into the embossing of the metal surface itself, the resistance of the metal to deformation through a movement of the die of an eighth of an inch would absorb all the energy of the 200 lbs. falling fifty inches, or 10,000 foot-pounds. If the resistance were uniform through the whole travel of one-eighth inch it would be equal to 80,000 lbs. pressure, but if the resistance increases uniformly from 0 to a maximum, the maximum would be 160,000 lbs. We would have to substitute for that drop press of 200 lbs. falling 50 inches, a hydraulic press capable of giving a pressure of 160,000 lbs., which would be rather expensive.

Mr. W. F. Durfee.—In this connection I am reminded of a statement made to me some few months ago by a party who has had quite a large experience in the manufacture of drop forgings. He said that he had demonstrated that the wear of the dies was in a great degree caused by the rush of the air from under the forging up along the flanks of the dies and along the sides of the

plunger that compressed the metal, and as the result of quite an extended series of experiments he had demonstrated that nickel-plating the dies was a very great improvement, and prevented in a very great degree the wearing of the dies from that cause. I am not able to state from personal experience the facts in regard to this matter, but as the prolonging of the life of dies for drop forgings is of importance, I take the liberty of calling the attention of those interested in drop forgings to this improvement.

No. 579—102.

What is the best mode for packing the rods and plungers of hydraulic machinery, particularly under high pressure?

Mr. John H. Cooper.—The earliest form of stuffing-box, which is reproduced in perhaps greater number than any other at the present time, is that in which the bottom of the box and the end of the gland are conical (Fig. 174), having the inclination toward the rod; the object being to gather the packing in from the outer circumference and force it against the moving part to secure close contact; the object of course being to obtain tightness, for preventing leakage of the fluid under pressure within the cylinder, from which the rod, plunger or trunk passes into the atmosphere.

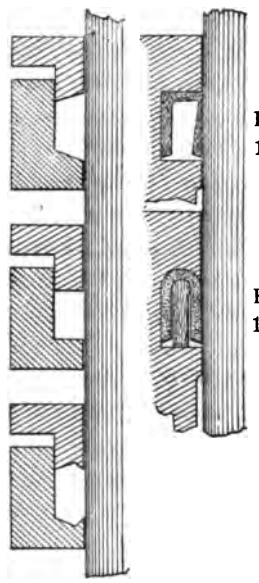


FIG. 174.

FIG. 175.

FIG. 176.

FIG. 177.

FIG. 178.

Equally as good results have been obtained by the use of square ends (Fig. 175) as by the bevelled ones. Such have been numerously made and put to the test of everyday successful service, but the writer is not aware of any comparative trials having been made to prove the better form. This has at least the advantage of economy of time in its production.

The effect of the bevel is to force the loose ends of fibrous packing into the joint space between rod, head, and gland, which in time must gradually wear them away. Naturally

enough a square end would have less tendency to do this, and ought to be a better form, if this reasoning be correct. It is fair to presume that it is. The writer was informed that a remedy had been found for this waste and consequent leakage under the higher hydraulic pressures.

This remedy is given in Fig. 176 of our ideal sketches. The bottom of the box and the end of the gland are grooved or concave, as if it were proven that the drifting of the fibres into the joint could be prevented by an arresting edge or flange; which has been done, and so Mr. Dickie of San Francisco says, "That's a good stuffing-box."

Many of us have used cupped leathers, both single and double, with, no doubt, good and bad results. I believe that much of the bad has been caused by using a low grade of leather, as well as by lack of support at the point where it bends to the moving part, and is perhaps subjected to greater punishment at that place.

In the double-cupped leather it is noticed that they wear mostly at this point, as shown in Fig. 177; but if, instead of making their chambers square-cornered, as is usually done, they were shaped to fit the curve of the leather, as in Fig. 178, and then held in place by a wooden ring, made in segments for easy insertion, they would last much longer. The writer has tried this plan, and can report favorably thereon.

The necessity of supporting the weaker members of a structure we are not slow in understanding; to apply the remedy, when known, is also within our business philosophy.

Mr. Erwin Graves.—The methods of packing for hydraulic machinery may be divided into two classes: First, that in which the pressure of operating liquid itself serves to expand the packing and cut off escape; and, second, that when the packing material is, by compression from an outside application of force, brought into a sufficiently close contact with the moving parts to prevent the escape of the pressure liquid. The cup leather is an illustration of the first, while the ordinary stuffing-box with fibrous packing the second.

The cup leather has serious drawbacks when applied to the commoner class of machinery, but on account of simplicity and neatness, with capability of preventing leaking under all ranges of pressure, and with a minimum of frictional loss, is, and will continue to be, employed in a variety of ways, particularly in the

finer class of hydraulic tools. It may not be uninteresting to give a brief summary of published results of tests for determining the friction with this method of packing. The most complete of these is that conducted in England by Mr. John Hick, whose experiments were with plungers 4-inch and 8-inch diameters, with leathers from $\frac{7}{8}$ inch to $\frac{3}{8}$ inch, and pressures from 443 to 6,375 lbs. per square inch. With the 4-inch plunger, the frictional loss in percentage of the pressure on the 4-inch area, of one thoroughly lubricated packing, was from 1.25 with 443 lbs. to 0.75 with 6,375 lbs., and was the lowest (0.50) at about 2,000 lbs., the average of the whole range being 0.70; with the 8-inch diameter plunger, under like pressures, the percentage was from 0.42 to 0.33, with the lowest (0.26) at about 2,500 lbs., and an average for all pressures, 0.316. It was found that the depth of the leathers—that is, the width of the rubbing surface—did not affect the friction; that this was practically the same whether the leather was $\frac{7}{8}$ inch or $\frac{3}{8}$ inch.

A series of tests on this same subject was conducted at the erection of the St. Louis bridge. The plungers here experimented with were 6-inch and 9-inch diameters, and the pressures very much less than the English tests, ranging only up to 273 lbs. With the 6-inch plunger, the percentage of friction was, at 60 lbs., 1.92, and ranged down to 1.06 at 273 lbs.; while with the 9-inch plunger, under like pressures, the range was from 1.64 to 0.90. Tests with leathers of different depths gave practically the same result, agreeing in this particular with those of Mr. Hick.

Mr. Hick concludes the subject of his tests of cup leather packing with observations, of which the following is the substance:

That the friction increases with pressure.

That for different diameters under the same pressure the friction of rams will be in the proportion of such diameters.

That the depth of the leather does not affect the friction.

For determining the friction he gives the following formula:

$F = D \times P \times c$, in which F = total friction of leather collar; D = diameter of plunger; P = pressure per square inch; and c = a coefficient, which he puts for well lubricated at 0.0314, and for badly lubricated at 0.0471.

In practice it is not likely that the working part of a machine would be in the perfect condition of an experimental apparatus; hence if I were called upon to use this formula I should double or treble the result it gave for the friction of a leather packing.

In the following cases I have employed leather packing in "U" form, and made in a simple way, for application to some horizontal acting presses with cylinder 30 inches and 48 inches diameter, and which gave most satisfactory results both as to lasting qualities and effectiveness. To have made these in the usual manner, from one piece of sole leather, would have been impracticable; the ones used were formed of strips of best quality of "harness" leather, cut about 1¾ inches wide, with pieces joined together by laps about 4 inches long, tapered so as to have the joint the same thickness as the balance. The laps were stitched together on one edge only, and the whole formed into a flat, belt-like ring, the diameter of the recess in the piston which it was intended to pack. To put in place, the leather was softened in warm water and made as pliable as possible, and then bent by hand into the "U" form as it was inserted into its place; this done, the hollow formed in the dish of the leather was packed loosely with strands of hemp so as to fill the space and hold the leather against the cylinder when pressure of water was not performing this duty; the whole was secured in place by a flat iron ring bolted against the piston. As an example of the wearing and working of such arranged leathers, I would state that in service the piston is moved forward at each operation from 8 to 12 inches, with water under a pressure of 15 lbs., and that the tightening up is then done with water at 325 lbs.; the cylinder pressure required for each operation varying from 100 lbs. up to that limit, but that such increase of pressure produces only a small forward movement of the ram—probably not over ¾ of an inch. Under such service leathers stand, without attention, from 30,000 to 45,000 operations of the press. I would add that the backward motion to piston is produced by suction of the exhaust, and that both sides of the piston are fitted with leathers, facing in opposite ways. The long life of these leathers is undoubtedly owing to the small motion of the piston while there is a pressure of any extent acting.

Coming to the subject of the stuffing-box and fibrous packing method, and taking up the question of friction as found by every-day practice, one can encounter a wide range of results

and can find cases where the friction is several times what it should be. Where such excessive friction exists it may be traced to one or more of the following causes—bad form of box, imperfect packing material, unskilful work in packing, unnecessary or uneven strain on gland, imperfect plunger, or, finally, bad lubricant.

Most establishments employing hydraulic power are provided with an accumulator, and there is not a better apparatus for simple and practical observation of the friction of a plunger than this. All that is necessary is to attach a pressure gauge to the water connection and note the pressure in the rising and falling of the accumulator; when rising there is the resistance (which resolves itself into pressure per square inch) due to the load plus the friction, and in falling the load minus the friction; or, in other words, the friction in pounds equals one-half the difference in pressure per square inch, multiplied by the area of the plunger in inches. It is well to keep in mind that a plunger with fibrous packing set to resist a certain pressure does not, on a less pressure, follow the rule laid down for cup leathers and have a less friction in proportion to such less water pressure. On the contrary, with packing outwardly remaining the same, the total friction or power necessary to slide the plunger through the same is probably greater with the surrounding water at low-pressure than at high; the reason for this being that pressure to a certain extent compresses the packing and forces itself in between that and the surface of the plunger, thus acting in a double way to release the packing pressure and to lubricate the plunger.

The following are observations of accumulators taken in their every-day operation without any special preparation, but which had originally been carefully packed.

Accumulator with 11-inch diameter plunger, loaded to give 320 lbs., showed a difference of 10 lbs. between up and down motion; this represents a resistance of 475 lbs., or 1.56% of the load.

An accumulator with $4\frac{5}{16}$-inch diameter plunger, loaded to give 1,600 lbs., showed a difference of pressure corresponding to 219 lbs. friction, or 0.93%. This last is quite as low as could be expected with a cup leather. These accumulators were packed with $\frac{5}{8}$-inch "square flax."

I do not think that any particular method of packing can be

named as the best for all circumstances; much depends on the service, surroundings, and nature of the machine. In certain classes, where the required motion of parts in contact is small and the pressure great, some of the forms of leather, "hat," "cup," or "U," will give results that cannot be attained by any other means, particularly as to small space, tightness, and light friction. For the stems and rods of hydraulic valves and such similar work, this method will usually be found the best.

Of the larger and heavier class of work, such as cranes, accumulators, etc., the stuffing-box and gland should be used, and for the packing my preference is strongly for braided "square flax." With the class of machinery named, when an externally accessible packing can be used I should never employ leathers.

In designing stuffing-boxes I see no reason for much variation in the size of the packing space, and should confine myself to one size, $\frac{5}{8}$ inch or $\frac{3}{4}$ inch. The depth of the packing recess can be from 4 inches to 6 inches according to pressure, and the following shoulder of gland, half as much. To pack, obtain the braided flax of the size of the space in the box, and cut to length forming a ring; care being taken in doing this, so that when inserted the ends do not overlap and form a lump, or that there is a vacant space at the ends; these rings to break joints and each to be separately and firmly pressed to place. When this is done and the gland down, the nuts should be set up very slightly, and then loosened off so as to be turned with the fingers—leaving them in this condition to be tightened as may be necessary after being put under pressure. In use, keep the gland square with the plunger, and limit the tightening to as little as necessary to reduce the small leakage to the limit allowable. The oil used for lubricant should be of a quality that will not "gum," but leave the surface bright.

The plungers in construction must be turned with care, a difference in diameter between different places that is hardly perceptible will show very plainly when put into use.

As a supplement to the foregoing observations on the subject of hydraulic packing, I would submit the result of some experimental tests made with a view of throwing some light on the question of friction of fibrous packing under varying hydraulic pressure. The opinion which I held was, that in the case of a rod or plunger packed with material of this nature, the friction was not greater, and probably was less, with the pressure on

than off. On this subject I have been unable, after diligent search, to find that direct tests have been made; the question, though, has been raised and a difference of opinion expressed; an instance may be found in the 1887 volume of the Transactions of the American Society of Civil Engineers, in the discussion concerning the Athens testing machine.

To make these experiments, an appliance at hand was utilized, and, while the depth of packing-boxes was not what would be usual in practice, it is not likely that with deeper boxes and more packing, results would be obtained relatively very different. The accompanying sketch (Fig. 179) shows the apparatus employed; the plunger is 9 inches in diameter, and the boxes were each packed with three rings of $\frac{3}{4}$ inch square braided flax ("Clinton brand"), and glands set up sufficiently tight, so that at 400 lbs. pressure the leakage was reduced to about ten drops of water per minute; no lubricant was used other than that contained in the packing. The piece acting as a cylinder and carrying the packing-boxes was held rigidly, and the plunger moved through it, the necessary power for doing this being measured by a spring balance applied to the long end of a lever which had a fulcrum on one end of the plunger; the motion given to the plunger being six inches. It will be seen that outside of the friction of lever bearings and that of moving the dead weight of the plunger, the additional power required is that caused by the friction of two stuffing-boxes. In the tabulation of results the dead load friction has been deducted, and the figures given are the averages of over one hundred tests in both directions of the moving plunger. With each test the averaged pressure shown by scale through the entire motion was taken, and the order of tests with relation to pressure varied, one series beginning with 400 lbs. and working down, and the next in the opposite way.

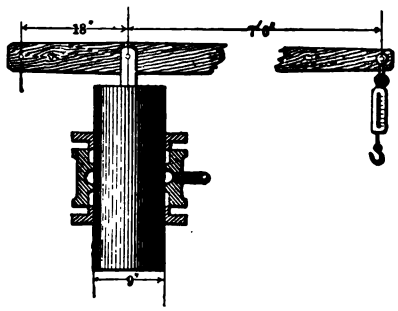


FIG. 179.

Friction of two stuffing-boxes —	no water pressure,	447 lbs.
" " " "	50 lbs. water pressure,	386 "
" " " "	100 " " "	367 "
" " " "	150 " " "	337 "

Friction of	two	stuffing-boxes	—	200 lbs.	water	pressure,	305 lbs.
"	"	"	"	250 "	"	"	275 "
"	"	"	"	300 "	"	"	275 "
"	"	"	"	350 "	"	"	267 "
"	"	"	"	400 "	"	"	263 "

Further tests with packing glands tightened to an unnecessary degree, while causing more friction, showed relatively the same differences with the varying water pressures.

Tests were also made for determining friction of rotating plunger. These were taken with packing in same condition as with above tabulation, and the power was applied on a lever 4 feet 6 inches from the centre of the 9-inch plunger. The results were as follows:

With no water pressure,				47	lbs.
" 100 lbs. water pressure,				44.5	"
" 200 "	"	"		40	"
" 300 "	"	"		38	"
" 400 "	"	"		34.5	"

The conclusion that with fibrous packing tightened up about a rod or plunger for resisting a certain pressure it offers more friction to the movement of the same as the pressure is reduced, appears to be verified.

Mr. Eckley B. Coxe.—At our mines, we have, in addition to the difficulties ordinarily incident to the packing of pumps, to contend with the effects of the sulphuric acid in the mine water, which attacks the plungers and renders it very difficult to keep the pumps in order. For the last five years, we have been using with success what I believe to be a novel system. The accompanying sketches (Figs. 180 and 181) show the plan adopted. *A-A* are the water cylinders, *B* is the plunger, and *K* is the cylinder which replaces the stuffing-boxes and contains packing. It is bored out so that the plunger will just slide through it. In the centre of it there is a large groove *L*, which is cast in the cylinder, having a hole *F* on top, into which is screwed a plug *C* and another at the bottom *G*, into which a piece of gas pipe is screwed. To the right and left of this large groove *L*, there are three small grooves *D* which are cut into the cylinder when it is bored out; and still farther to the right and left a groove *E*, which is similar to *D*, but wider. The pipe *M*, which is screwed into the cylinder at *G*, connects with the

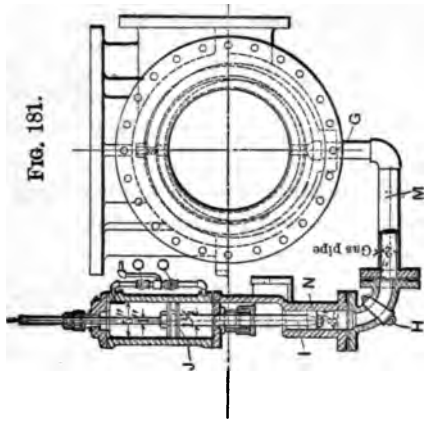


FIG. 181.

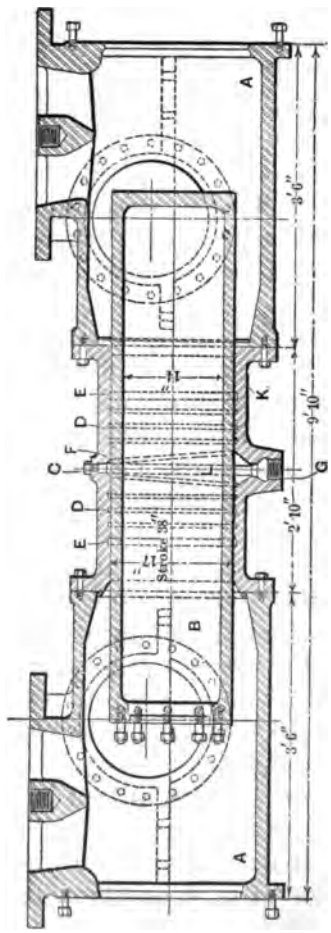


FIG. 180.

bottom of a small cylinder *I*, which is known as the grease cylinder, in which a piston moves which is actuated by a water engine *J* placed immediately above it.

The packing consists of beef-fat which is taken from around the kidneys. If you take a piece of this fat and try to tear it, you will find that it is full of fibres, and that it does not tear easily.

By means of the two stop-cocks *O O* the pressure of the water from the column pipe can be turned either above or below the piston of the water engine, and by the same stop-cocks, the water allowed to flow out of either end. By turning the water under the piston of the water engine, the piston *N* of the grease cylinder is raised out of the latter. This cylinder is then filled with the above-described fat cut into pieces about one inch cube, which have been thoroughly rolled in graphite. The water is then reversed, and this grease forced through the pipe *M*. This continues until the pipe and the large groove *L* are entirely filled with a solid mass of compressed grease, which finally leaks out of the hole *F*, looking very much like an ordinary candle. The hole *F* is then closed

with a plug, the grease piston *N* turned back and filled with fat, and the pump is ready for operation. Two piston rings are sprung in the two outside grooves *E*, which close upon the plunger and prevent any outflow of grease. It will be observed that the groove *L* is much wider at the bottom than at the top: this is so regulated that the excess of upward pressure on the grease carries the plunger and prevents it wearing the bottom of the packing cylinder more than the top. After this has been done, it is only necessary, from time to time, as the grease plunger descends toward the bottom, to reverse the water and fill up the cylinder. Practically, we do this every morning, as the amount of grease used in a day does not require the filling of the cylinder more frequently.

The plunger running through the grease under this pressure becomes thoroughly coated with the grease and graphite, so that a drop of water placed upon it assumes a spheroidal form and does not wet the plunger. The consequence is, there is no action of the acid water on the iron, except at the ends, which, of course, are made heavy enough to allow for a long-continued action of the water, although we find the ends in practice to become covered with grease so that the action is very slight.

The apparatus itself is very much like a sausage machine. The first one was put in in April, 1889, at a place where we had very bad water. The plungers here seldom lasted more than six months, and the same plunger we put in in April, 1889, is still running and has had nothing done to it. We have measured it once or twice, but find no appreciable wear. We have these pumps running under a pressure of from 200 to 400 feet of water, and have no trouble with them. It takes about one pound of grease per day to run a plunger 16 inches in diameter with a 6-foot stroke, making from 10 to 20 strokes per minute. This grease is worth about four cents per pound.

We have this apparatus now running with a large number of our pumps, and are putting them in as fast as we can get around.

In order to test its efficiency, we have often taken off the end of one of the cylinders, when the plunger was about beginning its stroke into the other, and allowed it to make its stroke, but there was no water visible.

One great advantage of this is that in our mines we are often obliged, during the heavy water, to run our pumps very fast, and

it is always at this time that the ordinary packing gives out, when the men have the least time to replace it, and have to do so under the greatest difficulties. With this packing it can be done in a few minutes, at any time, without interfering with the running of the pump. In packing the ordinary plunger pumps in a hurry, it is very doubtful whether the two piston stuffing-boxes are screwed up exactly even and brought exactly into line, in consequence of which very great additional friction is often brought upon the plunger.

Mr. C. W. Barnaby.—I would like to ask what difference, if any, you make in the relative diameters of the pistons there.

Mr. Coxe.—We first thought it necessary to make very accurate calculation so as to avoid too much pressure, but we now find it unnecessary. We now make our water cylinder larger than our grease cylinder, and always have a greater pressure on the grease cylinder than on the water cylinder, and we provide in the pipe a plug-cock *H*, which regulates the flow of grease and prevents any appreciable waste of it. We first tried to use tallow, but it would squeeze out. The fibre in the fat solved the problem.

It is important that the plunger should be turned as large as possible, consistent with its passing freely through the packed cylinder.

Mr. Kent.—Is the plunger-rod packed in the same way?

Mr. Coxe.—No, I have not gotten that far yet.

Mr. Kent.—Is it moved by a rod?

Mr. Coxe.—Yes, it is moved by a rod.

Mr. Kent.—Any trouble with corrosion in the rod?

Mr. Coxe.—No, we generally use bronze rods.

Mr. W. E. Crane.—Among the best packings is a kind called "Combination Packing" made in Cleveland, Ohio, and a square packing made up of canvas and rubber, but without rubber back, sometimes called "Tuck's."

A very important thing is the use of black lead. This is much better when mixed up in form of grease and afterwards thinned to proper consistency with oil.

The packing should be soaked, or rather covered with this, and rods and plungers covered with it when they are dry, as it will not adhere to a wet rod. In case of pump rods and plungers, some means must be provided to apply it continuously; but in case there are small holes or ports about any of the machinery,

it should be used judiciously or they will become plugged and closed.

On elevators, accumulators, and the general run of hydraulic benches, the application of the lead is not necessary oftener than once per day, and in many cases once per week.

For some kinds of work a moulded rubber packing after the general design of the ordinary cup packing of leather is excellent.

Mr. A. H. Raynal.—I have listened with great interest to this discussion, as at frequent times in my professional life I have been brought in contact with different classes and styles of packing.

FIG. 182.

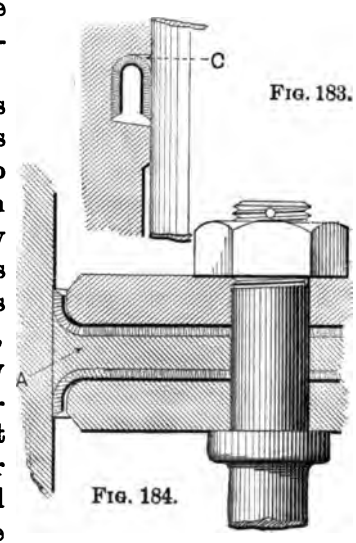


FIG. 183.

FIG. 184.

The hydraulic packing referred to in the paper by Mr. Cooper was well described in a paper read before the Iron and Steel Institute of England about 1870.

There are, however, some points of interest not mentioned in his paper which are of vital interest to us. It is with leather as it is with many other things, there are many kinds and qualities of leather. As we have money that is no money, as we have wood that is hardly wood, so we have leather that is hardly leather and unfit to be used in hydraulic packing. For instance, that part of the hide which is right over the neck and which is as hard and stiff as a brick. So first of all choose a good piece of leather, soft and pliable, yet strong; your common sense will soon tell you which to select.

Then again, like money, like the tariff question, like other things, leather has got two sides. This (Fig. 182) represents the thickness of a piece of leather. A certain portion of it, the upper part *A*, the hair side, is firm and solid. The other part, *B*, is made up of fibres running at right angles to it, having little cohesion and strength. This part *B* should be placed against the rubbing surfaces, for while the fibres will wear down, the part *A* will keep the fibres together. But were

you to place the part *A* in contact with the rubbing surfaces, then as soon as wear had consumed this small amount of thickness, all the rest of the fibres would lose their support and fall to pieces. Not enough care is exercised by the parties making the packings. As a rule, they are made by some handy "Jack." The superintendent hardly ever pays the slightest attention to it. Some "second-hand" clerk is asked to order the leather, and he buys what he can buy for the least amount of money. Then the handy fellow, sometimes the storekeeper, or the engineer, or even the watchman, sets about making the packing, never dreaming that there might be a difference which side is best for wear.

There is another point of great importance in hydraulic packings, and that is the manner of making the grooves which hold the packings, so as properly to support the rounding portion of the leather.

To illustrate, take the packing shown in Fig. 183, showing the usual groove in the cylinders of hydraulic presses. You will observe a small triangular corner space at *C*, which does not support the packing, and the friction of the plunger, which is great, has a tendency to crowd the leather gradually into this space, buckling the leather, breaking it, and wearing it out soon.

Fig. 184 shows portion of a piston for compressors with hydraulic packing, illustrating how the leather should be supported. Manufacturers of such pistons have learned to pay the greatest attention to the careful making of the distance piece *A* between the packings, making them of the very best material for the purpose of obtaining a sharp edge, generally using wrought iron, at times going so far even as to make the central portion of common iron, welding a ring of Swedish iron around the outside, turning it as sharp as possible, and even grinding the edges.

The lips of the leather are frequently made too long; they should be short, carefully cut at an angle, and a space should be left behind the leather, between it and the follower, to admit the fluid behind the leather to press it against the wearing surface.

Mr. W. F. Durfee.—Mr. Raynal has just touched on a point on which I wish to speak. In boring out the upper ends of hydraulic presses made of cast iron, it is exceedingly difficult to

get that thin cusp which is represented in the last cut on the right-hand side of Mr. Cooper's engraving, sharp enough. It is very liable to crumble and break away in the boring operation. That has probably caused the manufacturers of hydraulic presses to substitute a square groove, such as is represented at the top on the right-hand side, for the concave groove below. I have found it advantageous in such cases to put above the convex surface of the leather packing a ring of brass. This ring of brass is made in three or four sections, cut before it is turned up, so as to get it into place in case the press is made with a solid top. This ring is inserted just above the cup leather, and is turned concave on its under side. In that way the very sharp edge that Mr. Raynal has insisted upon, and very justly so, as being desirable, can be made without any difficulty. The construction works very well in practice.

Mr. Charles E. Emery.—The discussion has proved very instructive. There are three features I will mention which may be of interest in this connection. In regard to the reduction of friction with increase of pressure when ordinary packing is used, in stuffing-boxes, it may be remembered that for some years I have recommended, in making engine tests, that the stuffing-boxes containing flexible packing be slacked off before taking friction diagrams. I have taken friction diagrams with small engines which showed that the apparent friction of the engine was much more than the joint friction of the engine and load when tried with the dynamometer, which was traced to the use of flexible packing.

FIG. 185.

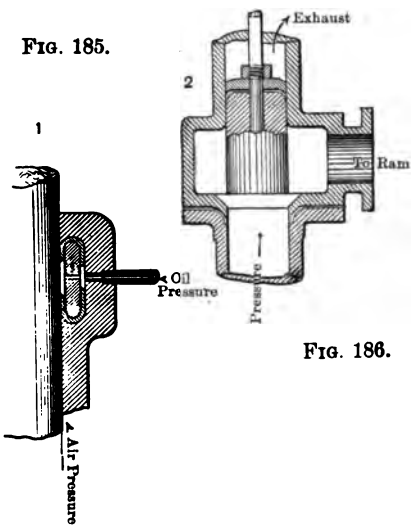


FIG. 186.

In the design of the dynamite guns, considerable difficulty was found in obtaining a packing which would be tight under a thousand pounds pressure. Ordinary leather packings were made in the various forms, and with all the care to get a sharp packing edge and other features spoken of here,

but they would persistently leak. Mr. George H. Reynolds finally insisted that the air went through the pores of the leather, and put in double cup leathers facing toward each other, and provided a pressure of oil between the leathers, which, as a whole, acted as a stuffing-box. (See Fig. 185.) The effect was that the oil under pressure kept the leathers out, stopped the pores, lubricated the rods and stopped the air leak.

Some here may remember an idea of mine in connection with reducing the apparent friction of cup leathers. For a hydraulic ram used as a straining press for a testing machine, I arranged to revolve the ram at a much greater velocity than the movement of translation when the two motions combined, exactly like pushing a fly-wheel along a shaft. We all know that if a loose fly-wheel be revolved, a man can push it longitudinally with his heel. The slightest force will move a weight one way, when the same is already in motion, in a direction at right angles to the motion desired. I reduced the apparent friction of the ram longitudinally in the same way; that is, by making the motion of revolution so great that very little difference of pressure was required in the other direction to overcome the friction. The plan was to have been adopted on the Government machine in case the A. H. Emery special features failed, which, of course, they did not.

There is still another point which will be appreciated by those who have had experience with cup leathers in elevator work. Some years ago, in attending to some work of this kind, it occurred to me that the idea of running the points of the leathers over holes was ridiculous, and I had some external cup leathers made, through which a plain piston entered against the pressure, and by means of external cup leathers, combined with leathers formed on the piston itself, a perfect valve was made, in which the pressure was always on the points of the leathers, without any holes whatever. (See Fig. 186.) You will find that in this way valves can be made very much more satisfactory than by the system of cup leathers riding over small holes. I probably patented the plan, but no more work of that kind coming in, the matter was not followed up.

Mr. Wm. Kent.—I would like to ask if any one can tell us how the plungers for those large compressing apparatus used in steel works are packed, such as at Cramp's and at Bethlehem, where these tremendous pressures are used for compressing steel.

No. 579—103.

Will there be any difference in the size of the chimney required, when the boilers are the ordinary tubular type, or of some of the water-tube forms?

Mr. Theo. F. Scheffler.—In my opinion, there should be a difference in the area, due to the difference in extra length of horizontal smoke flue; that is, the number of times the smoke travels the length of the boiler before getting to the chimney. Otherwise there should be no difference in the size of the chimney, other than the difference in the horse-power between a certain size tubular boiler and a certain size water-tube boiler, as the case may be. The area of the chimney in both cases should be determined from the quantity of coal likely to be consumed per hour per horse power, and by the difference of horizontal smoke flue. The height of the chimney being usually first determined, on account of local considerations, such as the by-laws of local authorities, the area at the smallest point of chimney may be found. When a tubular boiler has a third return flue over the boiler, the combined lengths of the three horizontal flues may be longer than the combined length of horizontal flues in a water-tube boiler of the same horse power. Where there is a greater length of horizontal flue, a greater chimney area is advisable. The proportionate effect of great length of external flues may be allowed for where a flue circuit of 100 feet is taken as unity, and a proportionate additional area of chimney must be allowed accordingly.

As an illustration, we will take a boiler of 80 H. P. and a coal consumption of 400 lbs. per hour, with a third return flue over boiler, and the combined length of horizontal flues 50 feet, and the stack 80 feet high. The formula for above would be as follows:

$$A = \frac{12\,C}{\sqrt{H}} \div 1.076.$$

Where A = area of chimney shaft in square inches at top or in smallest part;
C = number of pounds of coal consumed per hour;
H = height of chimney in feet above grates; and
1.076 = constant for horizontal flue 50 feet long; and
12 = constant.

Example:

$$A = \frac{12.400}{8.94} \div 1.076 = 498 \text{ square inches,}$$

which is the area of chimney at smallest part in square inches.

In the case of the water-tube boiler, if the horizontal flue was only 25 feet long, then the above would be divided by 1.152 instead of 1.076, which would give 465 square inches for smallest area of chimney for the water-tube boiler, making a difference of 33 square inches between the tubular and water-tube boilers. If we reversed the conditions—that is to say, the tubular boiler has the horizontal smoke flue 25 feet long, and the water-tube boiler flue 50 feet long—then the area for the water-tube boiler chimney would be 498 square inches and the tubular boiler 465 square inches. It will be observed by the above formula that if the conditions of both types of boilers are exactly the same, then the area of the chimney will be the same at the smallest point. But in the case of the tubular boiler in the first condition, the length of the horizontal flue was taken at the maximum length, *i. e.*, with the third return flue over the boiler; whereas the tubular boiler in ordinary practice only has two turns, which would amount to about 30 feet.

The writer cannot say for a positive fact that all water-tube boilers have a longer horizontal flue than the ordinary tubular boiler, but knows that on several different makes of water-tube boilers they are somewhat longer, but not enough longer to choke the draught if the chimney should be calculated without taking into consideration the length of the horizontal flue or the number of turns the smoke may have made in getting to the chimney.

Mr. W. O. Webber.—The size of the chimney would depend on the area of the flues, in the tubular type of boiler, and a larger allowance should be made for the tubular boiler, in regard to friction through the tubes, than for the water-tube type; but both should be figured according to the horse-power, and under the same conditions.

Mr. W. B. Levan.—I would like to say that this question of chimneys and boilers seems to be rather mixed. I was called in to look at four 100 H. P. boilers connected to a chimney 48 inches in diameter and 150 feet high. The maker of the boilers and the engineer in charge asserted that the chimney

had all it could do to supply those boilers with draught. When I looked the matter over I could not understand why the chimneys should be limited to those four boilers; and the result was, after a careful examination, I stated that they could put in four more boilers of the same dimensions, which they did. The chimney, according to the published tables, would only be available for 480 H. P., but up to this time boilers of 1200 H. P. are connected to it, so that the question of horse-power of chimneys as per published tables seems not to be understood. If we take the rules as laid down by Box, Peclet, Briggs, or Kent—Kent's horse-power tables come nearer to the results as shown by practice. He says 800 H. P.

I have found in my practice, extending over thirty years, that all the theoretical and scientific refinements are useless in determining the velocity of the draught and the power of chimneys. The circumstances and conditions are so complex and uncertain in actual practice that no theoretical formula can be expected to give results even approximately correct under all circumstances and conditions when in actual use, which are continually varying. To undertake and to determine the exact pressure or densities at the top and bottom of the chimney in order to arrive at the average pressure and density at a given barometrical pressure, especially when this pressure varies from day to day, is a useless waste of time.

The direction of the wind has a very decided effect upon the draught: when it blows strongly into the ash-pit of the furnace it may greatly increase the draught, and when it blows from the ash-pit it may lessen the draught, or the motive force of the preponderating column of cold air; also the condition of the atmosphere loaded with moisture, as well as the angle of chimney top, and many other uncertain circumstances. What we do know is, that if we wish to double the force of the draught of a given chimney, we must quadruple the height.

Mr. William Kent.—Water-tube boilers generally have a larger ratio of heating to grate surface than ordinary tubular boilers, consequently a smaller grate surface for a given horse-power, and require to have the coal burned at a greater rate per square foot of grate surface. This requires a greater force of draught, which may be obtained by a taller chimney. A plain tubular boiler with 4-inch tubes may give satisfactory results with a chimney 40 feet high, or even less, but a water-tube boiler of the

type in which the gases cross the tubes at right angles and are reversed in direction two or three times should have a chimney not less than 80 feet high. Of course for a given horse-power the taller chimney will require the less sectional area, the area approximately varying inversely as the square root of the height.

Mr. Eckley B. Coxe.—I want to say one word upon this subject. In fixing the size of the flue, what we should have in mind is to allow sufficient air to pass through the furnace to burn the coal. That is what is to be done. You should know the pressure under the grate and at the exit; in some of our boilers we have got to choke the draught, while in all of our old-fashioned cylinder boilers, which are not of the best style (we have put mud drums under them, and a water-tube boiler behind), we sometimes put a blower in the stack, which accomplishes what one of our members has done by using a suction fan and drawing air into the stack.

You want to get the velocity that will take all the products of combustion away and just enough to allow the boilers to absorb the heat that you develop in your furnace. It is also an important point for you not to have your draught so strong that you draw your carbonic oxide out before it has time to combine with the free oxygen mixed with the products of combustion. It seems to me there is no formulæ that will tell you how high your stack ought to be until you know how much power it takes to get the air from the furnace grate to the stack, and that varies with the construction and setting of the boiler. This is the reason I think all these calculations are not reliable.

Mr. W. B. Levan.—We have had this experience in Philadelphia where there were connected three 100 H. P. boilers to a stack 72 inches in diameter and 125 feet high. They could only get 260° temperature. It was re-lined, reducing the diameter to 42 inches, and they are now getting 420°. So this shows you can make a chimney too big as well as too small. These are facts, and they are stubborn arguments.

No. 579—104.

What is the best form of cylinder lubricator for engines carrying 140 pounds of steam pressure, or over?

Mr. W. F. Mattes.—It may be interesting to the members to have their attention called to some experiments made in this

matter of lubricating at high pressure. I refer to the importance of delivering cylinder oil into the steam pipe or steam chest in the form of a finely divided spray, instead of large drops or a small continuous stream, which has been repeatedly noticed by experimenters and has been commented upon in engineering publications and discussions. But for various reasons it has received very little attention in the way of practical application from either the makers or users of lubricating devices. One chief reason for this neglect has undoubtedly been the absence from the market of any satisfactory device for accomplishing a proper preparation of the oil, and another has been the general lack of definite information on the subject among engineers. In some preliminary experiments made several years ago, illustrated in the following figure, there is shown a steam pipe 7 inches in

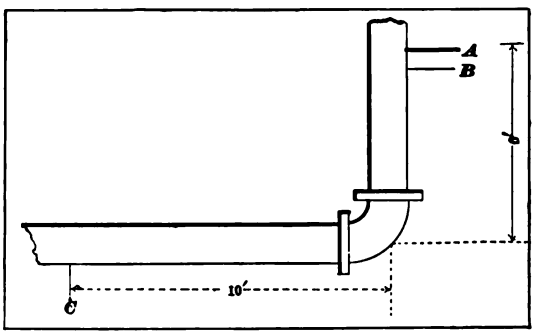


FIG. 187.

diameter leading from the boilers to a large compound pumping engine. On the vertical portion of the pipe, at positions shown approximately by A and B, were attached two lubricators, one of them being of a well-known type, in which the oil passes upward in large drops through water, and is then floated over through the connection to the main steam pipe. The other lubricator was an experimental design, arranged to spray the oil in steam before feeding into the pipe. On the under side of the horizontal portion of the pipe at C was tapped a small tube, running the upper end of the tube to within two inches of the top of the bore of the steam pipe, and connecting the lower end with a pet cock. Under the pet cock was placed a small piece of wrought iron, having the upper surface brought to an ordinary bright finish, with all greasiness carefully removed. Next, with the engine at

work and the cold-drop lubricator at *A* feeding in the ordinary course of business, the pet cock at *C* was opened for several minutes and a jet of steam allowed to impinge upon the bright surface of the wrought iron, after which the latter was examined in vain for any evidence of lubrication. On the contrary, the bright surface invariably rusted as soon as it dried. In sharp contrast with this result, whenever the experiment was tried by shutting off the cold-drop lubricator at *A* and starting the hot-spray lubricator at *B*, the iron surface would promptly gather grease and refuse to rust. The evidence was unmistakable that in the first instance the oil travelled sluggishly along the bottom of the pipe, oozing from every joint, while in the second it commingled promptly and thoroughly with the steam, and so necessarily reached every part of the engine touched by the steam.

Mr. Wm. O. Webber.—The best cylinder lubricator for engines carrying 140 lbs. of steam is the style of sight feed lubricator which is piped to side of the steam pipe above the throttle, and directly operated by the condensing and siphoning action of the steam, and preferably those where the drop of oil lifts through a glass tube similar to a water gauge.

No. 579—105.

What are the maximum safe speeds for hoisting and traversing in an overhead travelling-crane in a machine shop?

Mr. Wm. O. Webber.—We believe those in use by the Pond Machine Tool Co., on the Shaw electric crane in Plainfield, N. J., are probably the maximum safe speeds for hoisting and carrying, for overhead cranes. These we understand to be:

Longitudinal traverse	400 feet.
Greatest trolley traverse	200 feet.
Hoisting speed	50 feet per minute.

No. 579—106.

Is there any advantage in using a circulating device for the water inside a boiler, particularly of the three-furnace Scotch form?

Mr. John H. Cooper.—There is great advantage, because the boiling of water that lies below fire is a very slow process. The boiler plates are liable to be very unequally heated and expanded to do this, and are for this reason strained toward rupture, and made to leak often very badly at the joints.

Experience with this type of boiler has shown that the water lies cold and still beneath the furnaces, and serious leaks have occurred at the joints in this region.

To fire up one of these boilers from cold water to high pressure steam, without overstraining of the parts that lie in the vicinity of extreme changes of temperature, requires careful management and plenty of time.

It is just here that forced circulation of the water helps to promote and maintain the necessary equalization of temperature, by which strains are avoided.

An auxiliary boiler is sometimes used for heating the water in boilers of the type referred to, by transferring the heat of its steam to the water in its lowest part by means of an injector to which the name of Hydrokineter has been given.

This instrument is located, in a convenient water space, below the furnace tubes, and is also connected with the steam space of its own boiler, so that at all times the cooler waters in its lowest part may receive heat and motion from the steam above the water line, and thus maintain as near a uniform temperature as possible under the unnatural conditions of fire and water which exist in this type of steam generator.

The ordinary form of feed-pump may be used for the purpose of aiding circulation, by connecting its suction pipe to the shell of the boiler, a short distance below the lowest water line, and its discharge pipe to the shell, on a level with the bottom of the boiler at its coolest place.

Mr. A. H. Raynal.—I have tried the circulating apparatus of Mr. Wm. Craig, of this city, and found it very efficient. It consists of an injector nozzle placed in the feed-pipe between the feed-pump and the boilers, which picks up an induced current of water from the bottom of the boiler, discharging it with the feed-water in the upper part, thus creating a circulation, continuous while the boiler is being fed. This system is extensively used in New York harbor. The Pennsylvania Railroad Co. has it attached to the boilers of every tug and ferry-boat of their service.

I used it recently on the boilers of the U. S. cruiser *Bancroft*, and am convinced that the good circulation of the water in these boilers during the trials was an important factor in the excellent results produced.

Engineers in need of such devices will do well to investigate

this apparatus, and will, I am sure, obtain good results from its use.

No. 579—107.

Is there any best way to group the tubes over the cylindrical furnace of a Scotch boiler?

Mr. John H. Cooper.—The arrangement of tubes should be in vertical rows, with ample circulation space for water and nascent steam between them. Their attachments to the boiler head and "back connection," to which also the furnace tubes are secured, should be such as to give ample elasticity to accommodate the differing expansions, as it is well known, the upper part of the furnace, when the boiler is steaming, is longer than the lower part of the same, and the upper tubes of each group expand more than the lower ones; for whatever be the differences of length of these parts, due to difference of temperature, the plates to which they are fastened must suffer distortion, cause leaks at their joints, and become crystallized where molecular disturbance occurs from bending.

No. 579—108.

Will a forced circulation of air under a floor near the ground prevent decay of the wood-work and floor timbers?

Mr. W. W. Dingee.—I can report two cases showing that floors near the ground decay from lack of ventilation, and if the usual air-holes are insufficient for this purpose a proper supply of air must be furnished artificially, or the timbers will mould and rot. Some years since, in trying to season lumber in a tight brick kiln with ventilation at the top and steam pipes below, we found that the lumber would mould and the walls be streaming with moisture; but a current of air from a fan taking its supply from over the boilers prevented this and kept the atmosphere of the kiln dry. This experience leads the writer to place confidence in an artificial current for the purpose above mentioned.

The second example is in this city, where the lower floor of a brick building, 50 feet by 200 feet, erected two years ago, has rotted out. This floor was 3 feet above the original ground level, the foundation walls had the usual air openings, and after the building was erected these were protected, and the outside

grade raised to correspond with the floor. The joists were 3 inch by 12 inch Southern pine, and the floor a double one of 2-inch flooring, the upper being oak. There was a slight escape of steam from a pipe located below the floor, that kept the woodwork damp. New joists and the lower floor have been put in, and a current of air from a small fan discharged at different points under it, with the hope that this precaution will prevent a recurrence of this costly experience.

Mr. C. J. H. Woodbury.—The fungus growth known as dry-rot seems to be cultivated rather by the dampness than by the actual contact of water. I have seen timber used for bracing purposes against a ledge in the wheel-pit of a mill, where there were fissures through which flowed small streams of water. While the wet end of the timber remained sound, the portion of the timber back from the end rotted very freely. I have seen the under portion of the floor of a mill which was about two to three feet above the earth and where holes were left in the stonework of the underpinning. In those holes were laid 6-inch drain pipe as a finish, and there was a general ventilation obtained by the outside air passing through them into this low space under the floor, from which the air was exhausted by connecting the space with a picker room, where the blowers in the pickers forced air from the picker room to the dust flue, and this circulation of air at once stopped the sweating on the under side of the floors. When cold weather came on it was supposed that it would be very desirable to prevent the circulation of the outside air under the first floor, and the holes were closed with plugs. It was found in a very short time that the timber underneath was sweating again, and the circulation of air was resumed again and has been kept up to this day without any decay.

Another form of this question was, I believe, made a subject of a topical discussion at an early meeting of this Society. The question was whether a proper circulation of air would stop dry-rot. It appears to have been stated that if the generation of the germs producing dry-rot is once started, it is a very difficult thing to stop it. It does not appear to me that the mere change of condition possible to be applied to a building, such as the circulation of air, would stop it after that growth was once begun, although it might retard it. There is a method of treating wood, by one of the well-known preserving processes,

which consists in heating the wood to a temperature of about 350° by heated air under pressure in a closed drum, or chamber, and in that way driving out a slight part of the moisture; but it cannot all be removed from the wood, and, as there is no circulation of air through the drum, it merely throws the moisture and the gums from the inside to the outside of the cellulose cells forming the wood, and by that process it is stated that the wood is sterilized and is no longer a proper generating place for the growth of various kinds of mould and decay, and, in fact, each fibre of the wood is mummified. Whether this hypothesis is accepted or not as the correct one, it is well known that vulcanized wood will resist decay to an extraordinary extent. Even such woods as spruce, which is very susceptible to dry-rot under exposure to dampness, will last for a great many years, even in damp and swampy places The objections to most of the wood-preserving processes are that they consist of a solution which is applied to or forced into the timber, something which will only result in giving antiseptic properties until it is removed either by moisture or by exposure to the water, when the wood will rot quite as readily as previous to treatment.

Mr. W. F. Durfee.—It is within my knowledge that in a building erected in a Western city some few years ago, the roof timbers were left rough, and were cased up with pine boards painted to imitate oak, and the final result in an artistic point of view was very satisfactory; but after two or three years it was accidentally discovered that the whole of the timber work of this roof was so rotten as to be unsafe, occasioned simply because no proper allowance had been made for ventilation. It has always been my practice to leave an inch of space on either side of floor timbers where they enter walls, for ventilation. I have known instances where the timbers had become rotten and the whole of the floors had to be re-laid simply from neglecting to ventilate the ends of the timbers and thus keep them from sweating or fermenting.

No. 579—109.

Have you any facts to show that there is a molecular change in metals when their temperature is raised or lowered?

Mr. W. L. Cheney.—It has become the practice of at least one manufacturer of straight-edges, to draw them to a straw color

after hardening, instead of laying them away to season before finishing; this manufacturer claims that he thus accomplishes in a few minutes all that was formerly accomplished by months of seasoning, and that no further practical change takes place.

I have noticed in my own work that fine steel cutting tools, when being brought to a finish by filing, do not resist the action of the file nearly as much if they are heated to as high a point as will allow them to be handled.

Mr. W. B. Levan.—I think if you refer to the minutes of our Cleveland meeting you will find this subject was very thoroughly discussed in regard to springs there.

No. 579—110.

Have you any data as to earth-work dams which would suggest the factor of safety with which they have been built?

Mr. John M. Goodell.—George Stephenson is said to have remarked that in engineering more is to be learned from failures than successes, and in hydraulic engineering this is apparently too true. An examination of earthen reservoir and canal walls which have failed more or less completely shows that in most if not all cases the accidents were not due to errors in the statical design, but to faulty details. The physical properties of earth compel the adoption of cross-sections which are ample to resist any hydrostatic pressures that may be exerted, so the true factor of safety of an earth dam is hardly a measurable quantity. A consideration of a few failures will show this.

The disastrous failure of the South Fork dam above Johnstown, Penn., is an instance of a structure having a great factor of safety statically, which, however, twice gave way. It was completed early in 1853, and comprised what would now be called a rockfill section, 72 feet high, and 100 feet on its base, approximately, forming the down-stream portion, with the cross-section of a right-angled triangle. Against this was a core wall of slate rock 3 feet thick on top and 30 feet, more or less, on the bottom, while above this, and forming the up-stream portion, was an impervious section of good earth and puddle having a well-paved slope of two to one. This dam was examined after the last failure by James B. Francis, W. E. Worthen, M. J. Becker, and A. Fteley, as a committee of the American Society of Civil Engineers, and pronounced by them to be thoroughly built and

capable of resisting successfully the pressure of the water in the reservoir. Its factor of safety was very large, yet it failed twice at least—once on account of a defect in the foundation of the culvert, and the last time on account of an insufficient spillway. An insufficient spillway is also said to have been the cause of the recent failure of the Pecos River dam a few miles above Eddy, N. M., although I am informed that in this case there was ample way for any ordinary run-off, and that the wash-out was due to an exceptional rainfall or cloud-burst.

Another recent failure of an earth dam of more than ample statical security occurred a few months ago at Portland, Me. Here the dam or reservoir wall was constructed of good material well consolidated. An overflow pipe ran down through this embankment from the high water level on the inside to near the outer toe, where it was connected with a drain to a sewer. Frost probably loosened the concrete facing at the mouth of this pipe, so that water could flow down the exterior and thus soften and eat away the earth. The overflow pipe was not provided with cut-off walls, nor was there a core wall in the dam, consequently there was nothing to hinder the washout after the first leak began.

Although a few engineers are opposed to core walls, an occurrence at a little reservoir dam in Central Massachusetts convinced the writer of their utility. Early one spring this dam was discovered to be leaking badly just below the top. After the water had been drawn down it was found that the solid "frost-bitten" top had apparently been loosened by the ice, or from some other cause, so as to allow a thin film of water to pass between it and the top of the core wall, which prevented the percolation from extending to any serious depth. It did not take the engineer in charge very long to build that core wall up to within a foot of the top of the dam, as he is pretty certain that but for its presence the dam would have been washed out before the leak was discovered.

It would not be difficult to cite more instances where earth dams having a large statical factor of safety were imperilled by faulty details; this, however, is hardly in place in a brief discussion. In conclusion, I desire to call the attention of the Society to dam No. 5 of the Boston water-works, designed by Desmond FitzGerald, under the direction of City Engineer William Jackson. In this dam, which was fully illustrated in

the *Engineering Record* of November 4, 1893, is embodied the wealth of experience and knowledge acquired in the construction of a number of similar dams for the Boston Water Board, and the structure is therefore of much interest to hydraulic engineers.

No. 579—111.

Has any one tried sand-blasting as a method to produce a surface on cast-iron before nickel-plating?

Mr. W. L. Cheney.—This method has been tried by a concern in Meriden, Conn., and abandoned.

It is only fair to say that there were two factions in the factory where this was tried, one of which, it is reported, desired that the method should succeed, and *vice versa.*

No. 579—112.

Is there any better method than japanning to protect steel from corroding?

Mr. John T. Hawkins.—The little experimental experience in this matter which I have to offer tends to substantiate what the question somewhat supposes, rather than to point out anything better than japanning for the purpose named.

There are so many different conditions under which steel and iron are exposed to corrosive agencies, that, possibly, some others than those mentioned might exhibit japanning as less effective than it seems to have been under the conditions named; which, however, would seem to provide as great a facility for, or tendency to, corrosion as any that may be named in every-day practical operations.

In lithographic printing machines there is used a set of rollers for depositing a film of moisture upon the stone or plate. These rollers are made with a heavy metallic core, covered first with a comparatively thick, soft, felt-like fabric, and over this a more dense and thinner covering of similar material; their function being to absorb and hold considerable water, which of course would thus always come in contact with the metallic core.

Previous to my experiments, so far as I know, the cores of these rollers were made of iron or soft steel, and covered with a

sheathing of brass tubing; the ends being fitted with caps, tightly fitted also to the smaller diameters of the rollers adjoining the journals, and brazed or soldered to the tubing, so as to prevent access of the water to any part of the iron or steel; iron rust being detrimental to the lithographic process if it appeared upon the surface of the rollers. This was so expensive a method of constructing these rollers, that in order to cheapen them I determined to try electro-deposition of copper upon them to replace the brass sheathing, but found that unless deposited to so great a thickness as to make them more expensive, that when sheathed with tubing, the cores would rust beneath the copper, and in many spots and places pass through and appear upon the surface. I tried nickel-plating with still more indifferent results. I finally tried japanning them.

The japanned rollers have been in constant use about two and a half years without any signs of deterioration or cracking off of the japan—which two latter results I feared after long use—while not the least sign of rust has appeared, so far as I know.

In testing the specimens under the three plans named, they were laid in water and allowed to lie thus for various periods. With nickel deposited about $\frac{1}{100}$ inch thick, rust spots would appear within twenty-four hours, and with copper of about the same thickness within a week; but with the japanned surfaces a submersion of several months failed to show the slightest indication of corrosion or change of any kind.

It was my intention, in case japanning did not prove satisfactory, to try the several processes lately promulgated which are intended to protect iron or steel surfaces from corrosion by producing upon them a coating of some oxide of iron; but the japanning proved so entirely effective, and as it did not seem possible to replace it by any cheaper process, I concluded to let well enough alone, and can therefore say nothing as to those oxidizing processes.

No. 519—113.

Which is the best process to repair old files—sand-blasting, etching, or re-cutting? Do any of them really pay?

Mr. W. B. LeVan.—The best plan is to throw them away and buy new ones. I have had an experience of twenty years in buying

files, and have paid out a great deal of money in getting them cut, and I can say that it does not pay.

Mr. J. F. Holloway.—My experience is that the very best way to repair an old file is to sell it for a quarter of a cent a pound and buy a new one, the best you can get.

Mr. W. E. Crane.—Sand-blasting files is of no use, as it makes the tooth shorter, and simply raises an edge which has no durability. In re-cutting, the temper must be drawn, and the quality of steel used for files is such that it deteriorates rapidly when re-hardened.

Mr. Henry Binsse.—I have given attention to this question for a few years past; and I am convinced by my experience that it pays well to have files re-cut, provided that an economical grade of file be bought in the beginning. Economy, to be consistent, must commence by purchasing that file which yields the largest return for the outlay, and it does pay to re-cut this grade of file; but it is a waste of money, in my experience, to buy cheap, common files, and this is the grade which does not stand re-cutting.

I can make my meaning clearer by relating the practical illustration which opened my eyes to the difference in the efficiency of files. A little flat block of tempered steel had been worn slightly concave, and was sent to me to be filed flat. We had promised to finish it by the following evening.

The work was begun at seven in the morning, and, when I came to my office, my foreman reported that the first file, a new one, had given out. A second file followed; and, before noon, my foreman again reported that no progress had been made, and that the third file was dull. All three files were of the same size and cut, but of different, well-known makes.

At noon time, a high-priced file of a celebrated make was procured, and the workman began to use it at one o'clock. The job was finished before six; and this file was in first-rate condition, and was used for long afterward.

This experience led me to buy a few files, paying for them about double the price I had been used to, and I found that, besides performing from 20% to 30% more work in a time unit, they outlasted from two to three cheap files. It appeared clear to me that this increased efficiency of the man's labor would warrant the use of the dearer file, leaving the longer life as clear profit.

A good file must be made of good steel; it must be cut carefully with a sharp chisel; and it must be carefully hardened and tempered. Cheap files are made, necessarily, of cheap, common steel. A blunt chisel will cut more files than a correctly whetted, sharp chisel, for the sharp chisel gives out sooner; but the blunt chisel forces up the metal, injuring the steel and making a short, stubby tooth, with incorrect clearance angles. The steel has been weakened by the hammer-like blow, and is in no condition for service. In addition, cheap files are made by unskilled, cheap labor, which does not know whether it is making good files or bad ones.

Finally, cheap files are tempered by methods which give a large product at the cost of quality: frequently the files are too soft, frequently the teeth are burnt, and often the surface is warped, so that it bears in spots only, and not over the whole surface.

The chief argument advanced against re-cutting is that the low price of new files does not warrant it. It is my belief that this grade of file is the dearest kind which you can buy. Of course, this cheap file will not stand re-cutting. I have brought here two files which cracked in hardening. You will notice that the teeth are still soft, although the steel was heated high enough to crack it.

Files made of good material may be re-cut twice, sometimes oftener; and they are more adapted for this, as I found that the high-priced files are from 12% to 20% heavier than the others. Files of excellent quality can be had at about 50% off the American list; and they cost about 35% off the re-cutting list for re-cutting. I can find no important difference between the new and the re-cut file, and at these prices there is a saving of about 30% in favor of re-cutting.

No. 579—114.

What experience have you had with iron cores or chills for straight holes of three-quarters inch diameter to three inch diameter through one and one-half inch thick upwards?

Mr. F. H. Richards.—The question calls for a statement of experience with the use of "iron" cores, or chills, for straight holes. But since there appears to be a better thing for the purpose proposed, it may be in order to refer, under this head,

to the use of copper chills as a substitute for those of iron or steel. So far as the writer is aware, the use of copper for this purpose was first adopted several years ago by Mr. Henry Richards, a leading mechanic and inventor of New Britain, Conn., for nearly thirty years intimately connected with the development of one of the leading manufacturing establishments of that place.

Large numbers of copper chills, of various cross-sectional forms and of various sizes, have been used with extremely good results. The copper, being a good conductor of heat, not only quickly abstracts the heat from the metal in contact with the chill, but is itself rapidly expanded, so that on the cooling of the casting the copper shrinks more rapidly than the iron, and becomes entirely loose in the core when the casting gets cold. By reason of this peculiar action, the chills do not have to be driven out of the casting, but drop out freely.

There is, however, a limit to the use of copper chills, which limit as yet seems to be undetermined. Owing to its lower melting point, the chill is not well adapted for use in very heavy castings; also, the softness of the metal in some cases puts a limit to the adoption. There is, however, a wide range of work —especially when the castings are not too thick—in which the copper chills will be found a valuable improvement upon any other kind of cores. In the branches of work coming under my observation, the durability of copper chills has been at least four or five times as great as that of iron or steel chills of the same size and shape.

No. 579—115.

What methods do you use to determine the cost of manufacture of your products?

Mr. Louis S. Wright.—I would like to speak about some methods for determining the cost of manufacturing, which are used in a works where the products are iron pipe, miscellaneous foundry work—*i. e.*, castings weighing from 15 or 20 tons down —with the work of machine shop, wrought-iron tank shop, blacksmith shop, pattern shop, carpenter shop, stable, yard, etc.

The plant mentioned employs in the neighborhood of a thousand men. The methods used are such as entail careful records being kept in detail in each department of all work

done daily; these records being turned in each day to one department, where the cost in labor and material is figured up for each particular piece of work, to which are added certain percentages for general expenses.

The methods can probably best be described, as follows: Considering material first: all material ordered, of no matter what nature it may be, is ordered on printed forms, which pass through the hands of one person in authority, who sanctions the ordering of the material. Each order has designated upon it the department where it is to be delivered on its arrival in the works. These orders are written by a manifolding process, so that a copy is kept by the party ordering the material. After the sanction of the order, it is handed to the Purchasing Department, who rewrite it on proper blank forms, which are sent to outside parties who are to furnish the material. The original order is sent to the receiving clerk, through whose hands all material passes as it comes into the works.

The duty of the receiving clerk is to check over and deliver to the proper department all material as it comes into the works, receiving from the department when delivered a written statement that the material is as they specified in their order. All bills for material pass through his hands and are checked for quantities and disposition; *i. e.*, the work to which it is charged.

In many instances material coming in goes to the storehouse; the books of which are credited with the material as it comes in, and debited as it is given out.

The duties of the store-keeper are to keep records of all material as it comes into the storehouse, and charge off from his accounts all material as it goes out, charging it to the proper department or work for which it may be used; the account of material on hand being kept on cards indexed after the fashion of a card catalogue, one card being used for each article. On these cards are entered the material as it comes in, the date of its arrival, the quantities, the amounts delivered from the storehouse, and the date of these deliveries. At stated times a balance is struck, and the amount of any supply on hand of a given kind is determined. The store-keeper, having received previous instructions of the amount to keep on hand, issues orders to replenish supplies in accordance with these instructions, as the supplies are drawn from the storehouse. The

store-keeper makes returns at certain regular intervals to the cost department—hereafter to be spoken of—of all material delivered, designating the account to which it should be charged.

Each department in the works makes daily returns to the cost department of the time spent by the men in their work and of the material used by them. The method of doing this is in a general way, as follows: Each workman receives a written order from the clerk of the department for the piece or job of work which he is to do. That order contains all the information which is necessary for him to find the pieces of material on which he is to work, and the drawing which applies to it, etc. The time of day that this order is taken out by him is noted on the order. When the work is done the slip is returned to the clerk who issued it, and the time of finishing is noted thereon. In this way each separate item of work is accounted for and daily returned to the cost department.

The above arrangement applies especially to the departments where the workmen work at tools or at benches, which would apply to the machine shop, wrought-iron tank shop, blacksmith shop, carpenter shop, pattern shop and foundry.

Methods of a somewhat similar character are used throughout the yard and stable.

In the pipe foundries each workman has his task for the day, and no such system is necessary; the same task being done over and over again daily.

All of these daily returns are sent from each department to the cost department. Before sending, they are certified to by the foreman of the department; then pass through the hands of the time-keeper, who checks the time given thereon of each workman for his work, against the time which is indicated at the time office as time which the workman has been in the works.

These returns then pass from the time-keeper to the cost department, who sort them over, charging time and material to each job of work.

To give some idea of the cost of keeping records in this way, it may be said that for two foundries, one employing about eighty and the other a hundred men, one clerk is necessary. For a machine shop employing about a hundred and twenty-five men, two clerks, with an additional clerk when the shop runs at night. For a blacksmith shop employing about fifty men, one

clerk. For a wrought-iron tank shop with a hundred and twenty-five men, one clerk. For a pattern shop employing about twenty men, the foreman of the shop attends to the clerical work. For a yard force of seventy-five men, one clerk. For a stable with sixteen horses, a stable boss attends to clerical work. Pipe foundries require no clerks and have no returns of this kind made.

These clerks earn from seven to nine dollars per week. The cost department requires four clerks.

Mr. E. P. Bates.—The gentleman has given us two important factors of cost: they are material directly chargeable, and the labor. It seems to me there is another item that is quite as important, which is the capital invested in the business. I would like to ask him whether he treats that as entering into the cost of the products, or how he treats it. I have known of some old-time machine shops that never took the cost of the plant into consideration. They may have had $50,000 invested in real estate and machinery, and it would cost $500 to $1,000 to make yearly repairs. The proprietor may have had outside investments which brought him a revenue sufficient to maintain his plant. Under these conditions, if a casting was made which cost two and three-quarter cents per pound, and was sold for three cents, it was estimated there was a neat profit of a quarter of a cent per pound, whereas they were doing business at a loss. I would like to know how he accounts for the interest on his capital invested, and how much of a figure it plays in that class of work, assuming that the manufacturers occupied their own buildings and did not pay rent.

Mr. Wright.—In answer to the question, I may say that in looking at the running of your plant you have the land, the buildings, the tools, the furniture and fixtures, the labor, and the material. If the plant is a new one, it is probable that a part of the land bought may be considered as an investment. It is therefore set aside as such. The balance, together with its improvements and repairs to improvements, is devoted to manufacturing. On this land are buildings. Each building on this land is looked upon as having to earn for itself its proportion of the land cost; *i. e.*, it rents its proportion of the land, and therefore it is necessary to figure in the cost of operating expenses the cost of its proportionate value of the land, and in the same way each tool rents its proportionate part of the building on which it is located. In

this way is figured a percentage to add to each tool for operating expenses.

In looking further, the buildings will be found to be divided into productive and non-productive parts. The value of the non-productive parts is distributed as a percentage, and loaded on the cost for operating each productive portion. We now have our tools represented by certain figures in dollars and cents as rates per hour for their use; to these figures are added the rate of the employees operating the tool. These form the cost per hour for operating said tool or department. These figures, with percentages, are taken by the cost department, with the value of the material, and form the cost for manufacturing the product.

Mr. F. A. Halsey.—The various items that go to make the final cost of production may be classified, as follows:

Productive labor.
Non-productive labor.
Material.
Salaries.
Freight and cartage.
Fuel and light.
Supplies.
Insurance.
Rent (or interest and taxes).
Depreciation of tools and plant.
Repairs and miscellaneous expenses.

The cost of productive labor and material for any given product is easily determined and forms the basis of cost. The problem of finding true cost consists in determining the percentage to be added in order to cover the other items of cost named.

The total of each of these subsidiary items for any given period covered by a balance sheet can be accurately determined from the ledger in connection with the inventory, excepting only the non-productive labor, and the problem of finding true cost finally whittles down to the separation of the productive from the non-productive labor and determining the total of each.

My own method of doing this is to have every man whose name appears upon the pay-roll make out a daily time ticket covering his work for the day. These tickets are assorted day by day into the productive and non-productive classes, and the total

wages of each class for the day is determined. These totals are entered upon a page of the cost book kept for that purpose, and when a balance sheet is struck these items are totalled for the period covered by the balance sheet. Having these items determined, the others are taken from the ledger and the following items are added together:

Non-productive labor.
Salaries.
Freight and cartage.
Fuel and light.
Supplies.
Insurance.
Rent.
Depreciation.
Repairs and miscellaneous expenses.

The sum of these items gives the total loading for the period covered. The total loading divided by the total productive labor gives the percentage which the total loading bears to the total productive labor. The useful labor spent in producing any product, multiplied by this percentage, gives the loading which that product should carry; and the sum of useful labor, loading, and material gives the true cost. If desired, the loading can be figured and placed upon the sum of the productive labor and material, although I consider the method given to be the correct one.

If office expenses are considered as part of the cost of production, they may be included in the calculation; or, if the sales account is considered separate from the manufacturing account, these expenses may be omitted, in which latter case the result will be to give the cost of goods at the shop door.

No. 579—116.

Do you know anything about plaster of Paris and sawdust as a non-conducting covering?

Mr. C. J. H. Woodbury.—I think that the first mention of this combination of sawdust, flour, plaster of Paris, and coal ashes, as used in various combinations, including some or all of those four articles, first appeared in the correspondence in "Engineering," of London, a number of years ago, and was extensively copied into the technical papers in this country, and has been

used quite extensively. That mixture was tried in what were known as Ordway's experiments upon non-conductors of heat, which were submitted to this Society several years ago,* and which show that it was a very fair non-conducting material, and compared favorably with those that had a good standing on the market. It contains this difficulty, which is also true of all the materials which are put on in a plastic condition, and that is the injury in case of repairs, or of any accident by which the material is removed from the pipes. It is not nearly so durable as are other materials which are put on in shapes, fastened, and secured around the pipes. There is one element in the mixture requiring flour that deserves attention, and that is that the flour should be sour flour, the acid of which, with the alkali in the coal ashes, generates carbonic acid gas, which makes small cells throughout the mass.

Mr. W. E. Crane.—Sawdust and flour, equal parts, makes an excellent non-conductor, adheres readily to hot surfaces, and is easily applied. It is, however, combustible, and would take fire under high temperatures. Plaster of Paris is not a good non-conductor, but it is possible that, mixed with equal parts of the other two, it would become non-combustible and be a fair non-conductor.

Plaster of Paris mixed with sifted coal ashes loses its life and becomes dust when used for covering steam surfaces, and I should think the same objection would prove true with the above mixture.

Mr. W. F. Mattes.—I made several trials of plaster and sawdust some years ago at the Lackawanna Steel Works, and I reached the conclusion that it was a very fair non-conductor, but excessively unhandy in application.

No. 579—117.

In the formula for calculating thickness of boiler plate, what values should be assigned to the coefficient of reduction of strength due to rivet holes in steel? What is the best factor of safety in the same formula?

Mr. William Kent.—The ordinary formulæ for the thickness of shell, and for the allowable working pressure of a steam boiler,

* See experiments upon non-conducting coverings for steam-pipes, *Transactions* American Society Mechanical Engineers, Vol. V. p. 73 (October, 1883), Vol. V. p. 212 (May, 1884), Vol. VI. p. 168.

are $$t = \frac{f\,P\,D}{2\,T\,c} \quad \text{and} \quad P = \frac{2\,t\,T\,c}{f\,D},$$

in which t = thickness, D = diameter, T = tensile strength, c the coefficient of reduction of strength due to the rivet holes, f factor of safety. Assuming t, P, and D to be given, and T to be 50,000 lbs. for very soft steel, and 60,000 lbs. for stronger steel, what values would you give to c and f, for ordinary tubular boilers, horizontal slaws double riveted? Would it answer to specify 0.70 for c, and 5 for f with 50,000 lbs. steel, and 6 with 60,000 lb. steel? making the formulæ

$$t = \frac{P\,D}{14000} \quad \text{and} \quad P = \frac{14000t}{D}.$$

I would like to draw particular attention to the last formula and ask if there are any serious objections to it. The tendency of steel makers now is to get the tensile strength down near 50,000 for the very best fire box steel. That formula makes a very handy formula, no matter what the tensile strength of the steel is. The factor of safety varies with the tensile strength, as it should, the steel of higher tensile strength having the greater danger of brittleness. It corresponds to a working strain of 10,000 lbs. per square inch, whatever may be the ultimate strain. I would like to ask if any one would suggest another denominator than 14,000, and give his reason. It ought to be discussed by engineers what is the proper factor of safety, or the proper maximum strain to allow per square inch in designing boilers, and especially whether any greater working strain than 10,000 lbs. per square inch should be allowed.

Mr. C. H. Benjamin.—The results of a large number of experiments made in this laboratory on the tensile strength of punched steel plates give the following averages:

(1) Material, Otis steel boiler plate $\frac{1}{4}$ inch thick. Plates punched with ordinary flat punches showed a tensile strength 7.5% less, elastic limit 5% higher, and contraction 30% less, than a drilled plate of the same material, showing that the effect of the punching is to render the metal around the hole more brittle and less ductile than before. When a spiral punch was used, the ultimate strength was only 3% less than in the drilled plate.

(2) Material, a harder steel, $\frac{3}{8}$ inch thick. Plates punched with punches of various shapes and clearances showed a reduction of from 3.5% to 27% in tensile strength, the average loss being about 12.5%. In those tests, as in (1), the advantage of spiral and whistle shaped punches was very apparent.

Mr. W. B. Le Van.—I think it would be a good plan to go back to the old engineer's factor of safety, 6. The trouble is seen when elongation gets into the calculation. The factor of 6 might cover that point to a great extent.

Mr. H. B. Roelker.—The rules of the United States supervisors of steam boilers make the prescribed factor of safety for boiler shells about 4:1, or a little less. This has proven to be entirely satisfactory during twenty years' application, and I do not see why it should be increased in the case of boiler shells. The manufacturers of marine boilers would very generally protest against an increase.

No. 579—118.

What have been the recent directions of progress in the aluminium and other bronzes?

Mr. Leonard Waldo.—There has been considerable discussion and a growing interest for the past few years in the subject of aluminium bronze as material for structural engineering. It is not without a hope on my part, but it will have to be without a promise at present, that the results of recent experimenting and engineering in the casting and rolling and drawing of aluminium bronze may be put in proper shape for publication, either before this or some other scientific or technical society, so that engineers, who for professional reasons regard this material with favor, may have a statement in detail of the tensile strengths, elastic limits, limits of compression strains, permanence under varying temperatures, and the ductility and malleability of aluminium bronze.

It is a growing opinion among those who have watched the experimental development of aluminium bronze, that it is the most promising material since the time of the introduction of Bessemer steel, over which it has many advantages. It is unfortunately true that a great deal of thinking, and nearly all of the writing, on the subject of aluminium bronze has been based upon limited knowledge. In fact, it is open to question whether Mr.

Percy, in his first communication to the Royal Society on this subject, should have called the mixture of aluminium and copper aluminium bronze at all. He had produced a 5% alloy of aluminium and copper, which he at once recognized as a metal of beauty, strength, and prospective value. The union of aluminium with copper is attended with the evolution of great heat and large shrinking and a complete absorption of the aluminium by the copper. There is no subsequent liquation of either metal, and no commercial method of effecting a separation of the aluminium from the copper. I do not like to go so far as to say that the aluminium and copper have entered into chemical combination, but there are many reasons for thinking this has been the case, and I have seen no scientific evidence produced against it. It is certainly true that the most radical change has taken place in the quality of the compound resulting from the union of these two metals, as distinct from the qualities of either component. Possibly an aluminide of copper is formed which is soluble in molten copper, and if we accept this explanation we have a working theory which explains most of the observed phenomena in connection with the chemistry of aluminium bronze. This combination may take place according to varying atomicities of the aluminium.

It is a common remark among foundrymen that copper will absorb about 3% of aluminium. This is grown of their experience with the large shrinkage of the higher percentages of aluminium in copper. Such a low percentage of aluminium in copper results in very great ductility, but has no special value for structural work, except in the shape of forged or drawn metal. When you increase the percentage of aluminium up to about 9.5%, you have obtained a metal which still can be worked hot or cold with practically the mechanical results of the higher grades of steel. The tensile strength of such an alloy in castings of small size, and casting in either sand or chill moulds, may be as high as 90,000 lbs. per square inch, with an elastic limit of perhaps one-half this amount and resistance to compression strain of 150,000 lbs. per square inch, and a practical permanence of these qualities under temperatures up to twice the boiling point of water. As a matter of practical construction it would be interesting to you to know that we regularly fill specifications at our works for drawn bolts of this metal which shall stand a strain of 200,000 lbs. per bolt, of 1.75 inches sectional diameter, and

that in sand castings it is possible to guarantee the metal against seepage under several thousand pounds per square inch hydraulic pressure.

I ought perhaps to give a word of warning as to the difficulty of making aluminium bronze castings. The metallurgy of the metal has only begun to receive the attention it calls for to surmount the difficulties of making sound work. The rules of foundry practice in regard to copper alloy castings have very little application to the casting of aluminium bronze, and it would be a mistake, if you should fail to secure sound castings in your own foundry, to assume, therefore, that the problem possessed insurmountable difficulties, or that real progress had not been made, and was available in the present state of the art. The history of steel casting is familiar to most of you. I do not know that aluminium bronze presents difficulties less in number, though they are somewhat different in character, especially those relating to the melting heats and the precautions to be taken in pouring the bronze.

Had aluminium been discovered by the metal-worker, rather than by the chemist, the purity of the aluminium used would have been insisted upon from the start. In a large part of the work which has heretofore been done with aluminium in copper, enough impurity has been introduced by the use of impure aluminium to vitiate the working qualities of the copper itself; and a large part of the work, until quite recently, in aluminium bronze, therefore, has been unsatisfactory, because of this want of knowledge. At present our electrolytic metallurgists have brought the purity of the aluminium product to that state where 99% of pure aluminium can be insisted upon in buying for use; and with this purity, and the purity of Calumet and Hecla copper, bronzes of the utmost excellence can be made. Now, if it is true, as one of the papers claimed in discussing the price of copper, that it can be laid down in New York at a small profit at 6¾ cents per pound, and if it is also true that the modern cost of producing aluminium is approximately 28 cents per pound, it will be seen that a 10% aluminium bronze comes within the range of possibilities for large work.

As a matter of interest to the members of this Society, I have the pleasure of handing around little bars of aluminium bronze, just as they were taken from the sand moulds, and with one end put against the burnishing wheel, so that the color of aluminium

bronze may be brought out. You notice the finish of the surface bears the rough marks of the sand mould, and the metal is left so intentionally, in order that you may see the metal as it appears on leaving the sand. I call this, which contains about 10% aluminium and 90% of copper, aluminium bronze. Now, when this bronze, as you see it here, has an equal amount of copper added to it, by weight, we call that "half aluminium bronze;" and when this half aluminium bronze has an equal amount of copper added to it, we call that "one-quarter aluminium bronze;" and there is also a grade called "three-quarter aluminium bronze," which contains three-quarters as much aluminium as the brand aluminium bronze. We have made this classification of the bronzes to get rid of much of the existing confusion. The metal known as aluminium bronze contains approximately 10% of aluminium, and physical tests have shown that this unites in itself the best average combination of strength, ductility, and elastic limit. I should be glad if the use of this terminology became general. This is the first public announcement of it, and I think it is sufficiently clear, so that you will instantly comprehend the meaning of the term one-quarter aluminium bronze in technical literature, if it should be generally adopted. I venture also to hand to the members present some spoons of one-half aluminium bronze. I want to explain, in reply to possible questions, that they are not gold plated, and that nothing has been done to the metal except the ordinary processes in spoon-making, of stamping, swedging, and polishing. You will notice that they are undistinguishable from gold in color, and that they are stiff and strong. I am sorry that there do not seem to be enough spoons to supply the demand, and if any of the gentlemen present has sufficient interest in the matter, and will send me his address, I shall be glad to see that a specimen is sent to him. I have to thank you for the attention you have accorded me, especially as I have not yet the honor of being a member of your distinguished Society.

Mr. Holloway.—I move that the thanks of the Society be presented to Mr. Waldo for the very interesting remarks that he has given us upon the subject of aluminium bronze, and especially for the spoons.

This motion, being duly seconded and put to a vote, was announced carried by the chairman.

The Secretary.—I understand that it is designed that all these

samples of bronze should be retained by the fortunate men who get them. They are not to be returned.

No. 579—119.

AN IMPROVEMENT IN THE PLANIMETER.

There was exhibited by its designer an improvement in planimeters, in which the tracing point was replaced by a circular glass piece having two intersecting lines on the under side, and in explaining its use and advantages, Professor Jacobus said: "The instrument is shown in Fig. 188. The glass piece is at *A*,

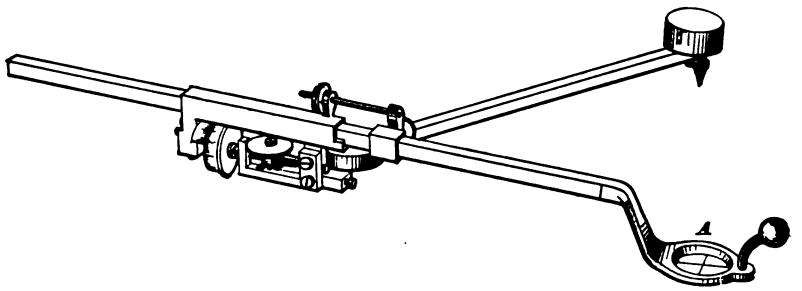


FIG. 188.

and the position of the cross lines, the intersection of which takes the place of the ordinary tracing point, may be plainly seen. These lines may be made short, so as to form a small cross, if desired, or other designs may be adopted to indicate the position of the tracing point.

"In measuring the area the intersection of the lines is moved over the diagram. The ordinary planimeter point has a tendency to trace a groove, and in going over the diagram a second time there is a tendency to follow this groove. Also on repeatedly going over an indicator card there is a tendency to obliterate the pencil line. Both of these disadvantages are done away with by the use of the glass disk. Another advantage of the glass disk is that it can be used in the evenings, whereas an ordinary pointer cannot with accuracy, on account of the shadow which it casts."

DLXXX.

GEORGE H. BABCOCK.

IN MEMORIAM.

For the first time in its history, the American Society of Mechanical Engineers is called upon to mourn and record the invasion of death into the ranks of those who have held the presidential office.

Mr. George H. Babcock, of the Babcock & Wilcox Company, who had served the Society as president during the term beginning December 2, 1886, to December 1, 1887, passed away December 16, 1893, at his home at Plainfield, N. J. It is a singular coincidence that his valued partner and life-long friend, Mr. Stephen Wilcox, also a member of the Society, should have been removed by death on November 27 of the same year.

Mr. Babcock was born at Otsego, N. Y., June 17, 1832. He inherited skill in mechanical invention from his parents, and made his first start in life in the woollen mill industry. While still a lad he started a printing office at Westerly, R. I., and founded a journal called the "Literary Echo," which is still a successful publication under the name of "The Narragansett." His tastes led him to interest himself in photography, and all through life he manifested great interest in the successive achievements of this art. His connection with printing brought him into contact with printing-press manufacture, and a partnership was formed between himself and Mr. Charles Potter, then treasurer of a corporation in Westerly, and for some years they were in the field developing this line of work. The polychromatic press for printing in several colors at one impression was the invention of Mr. Babcock, and upon this press and its method all press machines for color printing have been developed. The first printer's bronzing machine was also his invention.

He came to New York and entered the employ of Thomas D. Stetson, then one of the large practitioners of patent law, and his experience in this connection was of great service to him in later business undertakings. It was while in this relation that he

Geo. H. Babcock

taught mechanical drawing to a shop class in the Cooper Institute series, and many helpful suggestions are the result of his experience and labor.

He afterwards connected himself with the Mystic Iron Company and with the Hope Iron Company at Providence, as draughtsman, and it was while at this work in Providence, in connection with his ingenious friend and partner, Mr. Stephen Wilcox, that the design of the Babcock and Wilcox engine was developed and manufactured. It will be recalled that this was an automatic engine, one of whose principal features was the successful isochronism of its governor and its steam-moved cut-off valve.

This was secured by abandoning the idea of the governing balls being drawn downward by the force of gravity acting upon them. The balls remained in a horizontal plane and were separated by the acceleration due to centrifugal force from the axis of the spindle, but the tendency to come inward toward that axis was brought about by a weight acting upon the spindle through a bent lever. It can be seen by this adjustment that it was possible to have stable equilibrium between the centrifugal acceleration of the balls and the action of gravity upon the weight, only at the speed for which the governors were adjusted. The engine would therefore tend to come back to speed when the variations of the load occurred above and below the mean. The steam valves were of the gridiron type, with a cut-off valve on the back poppet whose motion was controlled by a cam which this governor adjusted according to the load.

This engine was manufactured both in Providence and Baltimore, Md., and elsewhere, and was successful as long as the early Corliss patents were in force; but as soon as they became public property, competition so reduced the prices of manufacture that this relatively costly type became an unprofitable venture, and the business was dropped. This took place in 1886 or 1887. Previous to this time, in fact as far back as 1856, Mr. Wilcox had been laboring over an inclined sectional boiler of water-tube type, and the ability of the two gentlemen who later formed the firm of Babcock & Wilcox for the manufacture of boilers was directed to the perfecting of a safety boiler of this design.

This early work was the basis of the Babcock & Wilcox Company boilers, and in 1868 both partners came to New York to push the manufacturing business.

It was at this time that the New York Safety Steam Power

Company was started for the building of small engines, for launch and other purposes, which might be operated in connection with safety boilers. The boiler business crowded out the other departments of the undertaking, until in 1878 it absorbed the entire time which the partners could spare, and in 1881 a stock company was formed of which Mr. Babcock was made president and in which his interests were large.

Mr. Babcock was one of the early members of this Society, identified with its formation, active in its committee work and on its Council, and its recognition of his service and ability culminated in his selection for the office of president at the close of the term of Mr. Coleman Sellers. He presided at the meetings held at Washington in May, 1887, and in Philadelphia in November of the same year.

His death was due to a kidney affection, which was met by a surgical operation which seemed the only possible step, and although he rallied after the operation, death resulted two days later.

Mr. Babcock was possessed of a quick, decisive disposition, able to bring to bear upon any subject all the knowledge he possessed in relation to it. He had experimented over a wide field of practical application, and was continually able to throw light from his experience upon any question which might be presented. His remarks in discussion on papers were always helpful and suggestive. He could readily and rapidly take up a new subject and master it thoroughly in all its details. When he was fifty-eight years old he took up the study of French and acquired it sufficiently to read, write, and converse in it. He was requested several times to be one of the expert lecturers before the students of Cornell University, and several of his papers presented in this way are models of care and forcible presentation of facts, experience, and opinion.

In 1873 he became a resident of Plainfield, N. J., and since 1885 has been president of the Board of Education in that city.

He was a leading member of that branch of the Baptist denomination which observes as a rest day the seventh day of the week, and conscientiously abstained from business obligations which might affect the freedom of his Saturdays.

Of an affectionate, genial temperament, he was beloved by all who came under his influence, and his death will be widely felt in a number of circles. He left a widow and one son.

PAPERS

OF THE

MONTREAL MEETING

(XXIXth),

JUNE 5th TO 9th, 1894.

DLXXXI.

PROCEEDINGS

OF THE

MONTREAL MEETING

(XXIXth),

OF THE

AMERICAN SOCIETY OF MECHANICAL ENGINEERS,

June 5th to June 9th, 1894.

LOCAL COMMITTEE OF ARRANGEMENTS.

H. WALLIS, ESQ., *Chairman.*
R. ATKINSON, ESQ.
K. W. BLACKWELL, ESQ.
H. T. BOVEY, ESQ.
GEORGE BRUSH, ESQ.
J. COX, ESQ.
G. E. CUNNINGHAM, ESQ.
A. DAVIS, ESQ.
J. DYER, ESQ.
R. GARDNER, ESQ.
E. P. HANNAFORD, ESQ.
R. HERSEY, ESQ.
T. C. KEEFER, ESQ.
J. KENNEDY, ESQ.
W. KENNEDY, ESQ.
W. LAURIE, ESQ.
E. MCDOUGALL, ESQ.
C. H. MCLEOD, ESQ.
P. A. PETERSON, ESQ.
F. R. REDPATH, ESQ.
P. W. ST. GEORGE, ESQ.
F. L. WANKLYN, ESQ.

J. T. NICOLSON, *Secretary*,
Engineering Building,
McGill College.

THE XXIXth Meeting of the American Society of Mechanical Engineers was convened in the city of Montreal, Canada, on the evening of Tuesday, June 5th. The hotel headquarters were at the Windsor Hotel, and the professional sessions were held in the engineering department of the McDonnell Engineering Building. The opening session was called to order by Mr. Herbert Wallis, chairman of the local committee, who introduced His Worship the Mayor of Montreal, J. O. Villeneuve, M.P.P., who read an address as follows:

To the American Society of Mechanical Engineers.

Gentlemen: I consider myself especially honored at being called upon to welcome to our city such an important body of men as the American Society of Mechanical Engineers, and that such an agreeable task should devolve upon me. I regard it as an honor paid not to myself, but to the entire population of the city which I represent. You have come to congregate in Montreal to hold the twenty-ninth meeting of your organization, and I trust that your deliberations will be as fruitful as your most sanguine hopes may expect, and that the esteem which we have already based on your repute will be deepened by personal knowledge into profound and strong attachment.

I cannot ignore the fact that one and all of you represent the great industries of the United States, and I am well aware of the importance of your body. The mechanical engineer has been the means of promoting the wealth, natural industries, and progress of every country in the universe, and of putting into economical and practical form the disposal of capital with the least possible outlay.

You are what may be properly termed the labor savers of the world; and the fallacy that machinery destroys the need of labor remains to be developed. I believe, on the contrary, that it has increased labor by increasing the output of all manufactured goods. The hard manual work of former days has been greatly improved upon, and it is men belonging to your call in life who deserve all the thanks and gratefulness of the people, as constantly using your energy and intelligence for the general weal.

I sincerely hope, that, during your sojourn in Montreal, you will give your attention to the industries we possess; and if you fail to learn anything new from them, we shall, nevertheless, reap the benefit of any improvements you may kindly suggest, and which we as a friendly people will be sure to appreciate.

On behalf of the whole city, I therefore tender to you a sincere welcome, and earnestly trust that your present visit will be happy and fruitful, and that you will long retain pleasant memories of Montreal.

I. O. Villeneuve,
Mayor of Montreal.

L. O. David, *City Clerk.*

Montreal, *June* 5, 1894.

Addresses were also read by John Kennedy, on behalf of the Canadian Society of Civil Engineers, in the following words:

ADDRESS.

The Council and members of the Canadian Society of Civil Engineers welcome most heartily the American Society of Mechanical Engineers, on this its first visit to Canada. The Canadian Society extends this welcome not only on its own behalf, but also on behalf of all the engineers of Canada. It does so, first, because it is a national society; and, secondly, because it uses the word civil in its original sense—as opposed to military—and the Society embraces within its membership, engineers in all branches of the profession. While in Canada our comparatively limited population still enables, or, perhaps, compels, us to concentrate our forces; your great numbers, combined with the ever-spreading tendency to specialism in engineering, as in all other professions, enable you to maintain great national societies, representing all the principal departments of engineering. There is, however, no branch of the profession which is not to a greater or less extent connected with the other branches, and particularly with mechanical engineering, of which, in America, your distinguished Society is the representative. Underlying all departments of engineering there are, however, the same general principles and a common foundation upon which all must build. There is also a community of interest arising from common tastes and from professional pursuits so closely related as to develop amongst engineers the world over a feeling of unity, which, although nominally divided, cements them in reality into one great whole.

We therefore again warmly welcome you as professional brethren, and also desire to express our high appreciation of the friendliness which has induced you, as a society of engineers of another nationality, to visit us and hold your annual convention in Canada.

On behalf of the Society,

C. H. McLEOD, *Secretary.*

The address of the Stationary Engineers of Montreal had been carefully illuminated, and was as follows:

TO THE PRESIDENT, OFFICERS, AND MEMBERS OF THE AMERICAN SOCIETY OF MECHANICAL ENGINEERS:

Gentlemen:—We, the humble representatives of Montreal No. 1 Canadian Association of Stationary Engineers, are here in behalf of that body to bid you a hearty welcome to our city.

Our feeling is, that we are related to you, in a certain sense, even closer than the proverbial "forty-second cousin," as from the hands of the mechanical engineers a large portion of your labor, skill, and inventive genius are handed over to the stationary engineers for economic operation, and to be maintained in that high state of efficiency in which it was received.

We welcome you because you are one of the most important and most essential bodies in existence. We welcome you because of the vast amount of valuable knowledge that will be disseminated in our midst—knowledge which will be of inestimable value to some of our citizens. We welcome you because this convention will give us still another opportunity of showing our friends from across the imaginary line, for no matter what the tariff may be, you will find our latch-string on the outside; and, further, there is no export duty on the friendly greetings which we beg you to accept.

You will most likely find that this city is behind the times in some respects, but in others we have no doubt you will, ere you leave, say we are in advance. You will, we are sure, find abundant proof that we are progressive, and intend to progress much faster in the future than we have in the past. Notice our large mills, our water works, our street railways, our immense harbor improvements, our grand university, and all in a city which is closed to navigation during nearly half the year.

It is with feelings of pride that we look over the list of professional papers you are about to discuss during this convention, embracing, as they do, some of the profoundest subjects in mechanical engineering, and of the most vital importance to the steam users of to-day. These papers will not fail to leave a lasting impression on the engineers of this city, an impression which in time will, no doubt, bear fruit in the more economic operation of our several steam plants, with a corresponding increase with profits to the owners; and, further than that, we feel that if it could be arranged for you to hold your convention here every year, this city would be benefited to an extent beyond measure.

In conclusion, gentlemen, let us again welcome, and wish you all enjoyment possible during your stay, and that we may soon see you here again.

Signed, on behalf of Montreal No. 1 Canadian Association of Stationary Engineers:

GEO. HUNT,
JOHN J. YORK,
THOMAS RYAN.

Montreal, June 5, 1894

Sir Donald A. Smith, LL. D., Chancellor of the University, and Prof. Henry T. Bovey, LL. D., welcomed the Society to the University and to the engineering building and laboratories. At the close of these addresses, President Eckley B. Coxe of the Society gave fitting response of appreciation, and delivered as an address the paper on *Technical Education*, which forms one of the papers of the meeting. The laboratories of the building, with their new and very complete equipment, were open and lighted for the inspection of visitors at the close of the session.

The hospitality of the city had not permitted the guests to feel neglected during the afternoon preceding this opening session, but the mayor of the city and the aldermen had invited the Society to be their guests on a drive to Mont Royal Park. Carriages conveyed the party through the picturesque foliage of the ascent, and up to the Belvedere, where a lunch was spread and appreciatively enjoyed. A suitable vote of thanks was passed by the Society at a later session.

Second Session. Wednesday, 10 a.m.

The report of the tellers of the Council on the result of the election of new members was presented as follows:

REPORT OF THE TELLERS OF ELECTION.

The undersigned were appointed a committee of the Council to act as tellers (under Rule 13) to scrutinize and count the ballots cast for and against the candidates proposed for membership in the American Society of Mechanical Engineers, and seeking election before the XXIXth Meeting, Montreal, Canada, 1894.

They have met upon the designated days in the office of the Society, and have proceeded to discharge their duty. They would certify, for formal insertion in the records of the Society, to the election, under the rules, of the persons whose names appear on the appended list to their respective grades.

There were 513 votes cast of the pink ballot, of which nine were thrown out because of informalities, the members voting having failed to endorse the sealed envelope.

Chas. W. Hunt, } *Tellers.*
Wm. H. Wiley, }

There were 466 votes cast of the yellow ballot, of which 20 were thrown out, the voter not having affixed his name thereto.

Wm. H. Wiley, } *Tellers.*
C. W. Hunt, }

Honorary Member Elected.

Isherwood, Benjamin Franklin.

MEMBERS ELECTED.

Atterbury, Wm. Wallace,
Blackall, Robert C.,
Blackburn, Arthur Henry,
Boyer, Francis Howard,
Brewster, Frank H.,
Brock, Wm. M.,
Carney, Charles Joseph,
Clement, Lewis M.,
Coffin, Wm. Carey,
Eldridge, A. H.,
Emory, Fred'k Lincoln,
Frith, Arthur J.,
Field, Burr Kellogg,
Flint, Bertram Pierpont,
Horstman, Henry J.,
Heffernan, John T.,
Hart, Chas. Edwin,
Herter, Emil,
Holly, Edgar Prentice,
Hirt, Louis Joseph,
Hunter, James D.,
Jenkins, Matthew C.,
Keller, Emil E.,
Kruesi, John,
Logan, Frank J.,
Langlotz, Charles,
Mantine, Charles F.,
Murray, Thos. Edward,
McMannis, Wm.,
Pierrez, Jesse C.,
Pascall, Richard Henry,
Piez, Charles,
Rider, Thomas J.,
Setarnes, Wm. Ellison,
Stieltjes, Fred'k,
Seavey, John Franklin,
Smith, Harold Babbitt,
Stewart, Reid T.,
Schaum, Otto Walter,
Souther, Henry,
Spalding, Chester M.
Stokes, Samuel E.,
Upton, Wm. Bailey,
Vanclain, S. M.,
Waldo, Leonard,
Wright, Jas. Knox,
Yereance, Wm. Burnett
York, Herbert Waldo.

ASSOCIATES ELECTED.

Frazer, Fred,
Glenn, H. F.,
Peckham, Wm. M.,
Russell, Edmund Allen,
Richardson, Clifford.

PROMOTED TO FULL MEMBERSHIP.

Bang, Henry A.,
Dixon, Walter Frank,
Foster, Chas. E.,
Foster, Ernest Howard,
Guthrie, Edw. B.,
Rowland, Chas. Bradley,
Rowland, George.

JUNIORS ELECTED.

Bowman, Franklin Meyer,
Burr, Jonathan Sturges.
Bramhall, John Asbury,
Bierbaum, Chris'r Henry,
Bunnell, Sterling H.,
Cruikshank, Lyle B.,
Dale, O. G.,
Guelbaum, David,
Hoffman, Jas. David,
Hale, Robert Sever,
Harvey, George Roy,
Kuhn, Joseph,
Kales, Wm. Robt.,
Logan, John Wood,
Merton, Jas. Millar,
Manning, Richard F.,
Miller, Alton S.,
Sangster, Wm.,
Smart, Richard A.,
Whitney, Alfred R., Jr.,
Wilkinson, Thomas L.

The following announcement was also read.

THE SIXTH INTERNATIONAL CONGRESS ON INTERNAL NAVIGATION.

To the American Society of Mechanical Engineers.

Attention is again called to the opportunity to join this Congress and secure its papers and the discussions on them.

The questions to be discussed were very fully mentioned in the bulletin of the American Society of Civil Engineers of March 15, 1894. They cover the construction and navigation of canals, the plant at ports, the means of preventing ice blockades,

traction and propulsion, the question of tolls, the relation between the configuration of rivers and depths of their channels, and the regulation of low water channels.

As these congresses are supported by governments of the countries in which they are held, the charge for membership is merely nominal, namely, five dollars, which may be sent to the Secretary of the American Society of Civil Engineers by those desiring to either attend the Congress or secure the papers.

The Congress will open on July 23d, and will last one week. During this time and shortly after, five excursions will be made to localities of interest about Holland.

The American Line offer round trip tickets on their steamers for $144. The last sailing which will permit attendance at the opening of the Congress is on July 11th.

The papers will be published in English, French, and German.

A meeting of the British Association for the Advancement of Science will be held at Oxford, commencing on August 10th, and members visiting Europe will have an opportunity to attend both functions.

The President announced the appointment, under Article 31 of the Rules, of a committee to nominate officers for the Society for the ensuing Society year. He presented the following names:

Mr. Worcester R. Warner.....Cleveland, O.,
Mr. B. H. Warren.....Stamford, Conn.,
Mr. R. J. Gilmore.....Providence, R. I.,
Mr. C. W. Nason.....New York City,
Mr. J. F. Holloway.....Cuyahoga Falls, O.,

and requested confirmation, which was given.

Notice was also given in writing, under Article 45 of the Rules, of amendments to the Rules, to be considered at the Annual Meeting in December. These amendments affect Articles 2 to 14, Articles 18, 26, 32 and 42.

Professional papers were then taken up as follows:

Notes on "The Theory of Shaft Governors," by A. K. Mansfield, received discussion by Messrs. Lanza, Rites, Jacobus, Ball, Baker, and Webb.

"Heat Units and Specifications for Pumping Engines," by A. F. Hall, was discussed by Messrs. Hutton and Cary.

"A New Recording Pressure Gauge for Extremely High Ranges of Pressure," by W. H. Bristol, was discussed by Messrs. Henning, Holloway, Laforge, Cary, Almond, Ball, McBride, Baker, Webb, Halsey, and Stetson. Much of this debate covered the treatment of steel springs outside of their use in steam gauges.

Note on "Compressed Air," by Frank Richards, was discussed by Messrs. Jacobus and Goodale.

"The Relation of the Drawing Office to the Shop," by A. W. Robinson, was discussed by Messrs. Capen, Randolph, Wright, Binsse, and Shaw.

The session then adjourned.

The Secretary's register at headquarters showed the following members in attendance during the convention :

Almond, T. R.,
Almy, Darwin,
Angus, Robert,
Ashworth, Daniel,
Babbitt, Geo. R.,
Baker, C. W.,
Ball, F. H.,
Barrus, Geo. H.,
Beach, Giles,
Beach, C. S.,
Binsse, Henry,
Bird, W. W.,
Boenig, Robt. W.,
Bole, Wm. A.,
Bourne, S. N.,
Brewster, F. H.,
Brashear, J. A.,
Brock, W. M.,
Brown, R. S.,
Blackburn, A. H.,
Burchard, A. W.,
Bulkley, H. W.,
Cadwell, W. D.,
Camp, Geo. E.,
Cary, A. A.,
Cawley, Frank,
Cheney, W. L.,
Christie, W. W.,
Colwell, A. W.,
Corbett, Chas. H.,
Coxe, E. B.,
Dale, A. G.,
Darling, Edwin,
Delancey, Darragh,
Edwards, V. E.,
Dick, John,
Engel, L. G.,
Fairbairn, W. U.,
Field, C. J.,
Flather, John J.,
Forbes, Wm. D.,
Freeman, Jno. R.,
Frevert, H. F.,
Galloupe, F. E.,
Gillis, A. R.,
Gilmore, R. J.,
Goodale, A. M.,
Greensmith, J. E.,
Griffin, C. L.,
Hall, A. F.,
Halsey, F. A.,
Hand, F. L.,
Harte, C. L.,
Hayward, F. H.,
Heggem, C. O.,
Henning, G. C.,
Holland, Jno.,
Hollingsworth, S.,
Hollis, Ira N.,
Holloway, J. F.,
Humphrey, Jno.,
Hunt, C. W.,
Hutton, F. R.,
Hyde, C. E.,
Ives, A. W.,
Jacobus, D. S.,
Jones, Washington,
Keeler, E. E.,
King, C. C.,
Knight, A. F.,
Kirby, F. E.,
Kirchhoff, Chas.,
Laforge, F. H.,
Lanza, Gaetano,
Loring, C. H.,
Longnecker, C. K.,
Low, F. R.,
Logan, John D.,
Mayo, John B.,
Marble, H. M.,
McBride, Jas.,
McDuffie, C. D.,
McClelland, E. S.,
McKee, J. J.,
Manning, C. H.,
Mansfield, A. K.,
Mason, W. B.,
Melvin, D. N.,
Merrill, A. L.,
Miller, F. J.,
Moore, D. G.,
Nason, C. W.,
Nicolson, J. T.,
Reed, H. A.,
Redwood, I. I.,
Rice, R. H.,
Rites, F. M.,
Robb, D. W.,
Roberts, Wm.,

Roberts, T. H.,	Scott, Olin,	Tabor, Harris,
Robinson, J. M.,	Sharp, Joel,	Tatnall, J. E.,
Rockwood, Geo. I.,	Shaw, A. J.,	Thomson, John,
Roelker, H. B.,	Simpson, Wm.,	Veit, R. C.,
Rogers, W. S.,	Sinclair, G. M.,	Warren, B. H.,
Roney, W. R.,	Smith, C. P.,	Watson, Wm.,
Ross, B. L.,	Smith, G. H.,	Webb, J. B.,
Russell, C. M.,	Snow, F. M.,	Webster, J. H.,
Sangster, Wm.,	Stetson, Geo. R.,	Weeks, G. W.,
Scheffler, F. A.,	Stiles, N. C.,	Wiley, W. H.,
Schoenborn, W. E.,	Swasey, Ambrose,	Waldo, Leonard.
Schutte, Louis,		

Total..... 133.

There was also a considerable number of ladies in attendance.

For the afternoon of Wednesday a special train over the Grand Trunk Railway conveyed the party with their hosts to Lachine, where the steamer *Filgate*, furnished by the courtesy of the Harbor Commissioners, conveyed the party down the river, through the Lachine Rapids, and then for a sail on the quiet reaches of the river below the city, to view the work of the Commission in progress, including the guard pier. Afternoon tea was served on the boat, and the landing was effected at the foot of the Lachine Canal.

In the evening a most enjoyable reception was tendered to the members and their ladies at the home of Sir Donald A. Smith, the Chancellor of McGill University. The house, the conservatory, the picture gallery, and the assemblage were all elements in the pleasure of the party.

Third Session. Thursday, June 7th.

Professional papers of this session were:

"The Theory of the Steam Jacket," by R. H. Thurston, and discussed by Messrs. Hale, Hall, and Rockwood.

"Results of Experiments with a Fifty H.P. Engine to determine Influence of Compression on the Water Consumption," by D. S. Jacobus, and discussed by Messrs. Ball and Barr.

"Cylinder Proportions for Compound Engines from their Free Expansion Losses," by F. H. Ball, and discussed by Messrs. Jacobus and Rockwood.

"A New Method of Compound Steam Distribution," by F. M. Rites, and discussed by Mr. Rockwood.

"Test of a Small Electric Railway Plant," by Jesse M. Smith and discussed by Mr. Baker; and the paper by W. S. Aldrich, on "Power Losses in the Transmissive Machinery of Central Stations."

The session adjourned for a visit and luncheon at the power-house of the Montreal Street Railway Company, after which a lawn party was given by Mrs. H. T. Bovey and Mrs. Frank Redpath upon their attractive grounds.

Fourth Session. Thursday Evening.

The papers of this evening were by M. P. Wood, on "Rustless Coatings for Iron and Steel," discussed by Messrs. Holloway, Henning, and Nason.

Jas. McBride, on "Corrosion of Steam Drums," discussed by Messrs. Hutton, Hawkins, Randolph, Davis, Grimm, Flather, Holloway, Babbitt, and Nason.

C. W. Hunt, "A New Mechanical Fluid," discussed by Messrs. Rogers, Rockwood, and Brashear.

F. R. Hutton, on "First Stationary Steam Engines in America," discussed by Messrs. Durfee, Binsse, and Woolsen.

John R. Freeman, on "A New Form of Canal Waste Weir."

The session adjourned in good season to witness a beam test in the University laboratory.

Fifth Session. Friday, June 8.

The papers of the concluding session were as follows:

"The Cost of an Indicated Horse-Power," by De Courcy May, and discussed by Messrs. Webber, Cary, Holloway, and Harte;

"Effect of Varying the Weight of the Regenerator in a Hot-Air Engine," by G. W. Bissell, and discussed by R. C. Carpenter;

"Mechanical Draft for Boilers," by W. R. Roney, and discussed by Messrs. Cary, Coxe, Holloway, and Rockwood;

"The Saturation Curve as a Reference Line for Indicated Diagrams," by R. C. Carpenter;

"Results of Measurements of the Water Consumption of an Unjacketed 1,600 H.P. Engine," by J. E. Denton, D. S. Jacobus, and R. H. Rice, discussed by Messrs. Ball, Rockwood, and Wright; and,

"Notes on the Corrosion of a Cast Steel Propeller Blade," by F. B. King, discussed by Messrs. Ball, Roelker, Henning, Coxe, and Flather.

The following resolutions were then passed with acclaim, and by a rising vote :

To the Governors of McGill University:

The American Society of Mechanical Engineers beg leave, through a unanimous vote of the members assembled in convention at Montreal, to express their thanks for the opportunity given to hold their sessions within the halls of the University, and for their kindly action in favoring the objects for which the Society has convened in the city of Montreal.

The American Society of Mechanical Engineers desires to present to the Mayor and Aldermen of the city of Montreal their most hearty thanks for the drive tendered to the members of this Society and their ladies to Mont Royal, Royal Park, and seek to express in this way their sense of recognition for the courtesy of the city extended to them.

The American Society of Mechanical Engineers wishes to convey to Mr. Herbert Wallis, Chairman of the Local Committee, their thanks for his capable presidence at their opening session, and for the arrangements which the Local Committee in charge of these matters made for the details of the meeting.

To Prof. John T. Nicholson, Secretary of the Local Committee, the Society would add its further appreciation of his labor, which has been so thoroughly and capably performed.

The American Society of Mechanical Engineers desires to express to the Grand Trunk Railway their thanks for the courtesy of a special train conveying the members and their ladies to Lachine.

It gave the members of the Society great pleasure to avail of this courtesy, and they seek in this way to give expression to their recognition.

To the Harbor Commissioners of Montreal, the American Society of Mechanical Engineers desires to express their sincere recognition and appreciation of the opportunity to visit Lachine Rapids under their guidance, and for their entertainment upon the steamer *Filgate;* for the opportunity to inspect the work of the commission which has been so serviceable in the development of the city of Montreal, and to examine the work in progress for the further improvement of the water front.

The Society seeks in this manner to return its hearty thanks for all these courtesies.

The American Society of Mechanical Engineers desires to express in this way to the Hon. Sir Donald A. Smith, LL.D., Chancellor of McGill University, their sincere appreciation of his words of welcome on the occasion of their opening session in the MacDonald Building, and to add an expression of their recognition of his interest in engineering education, and the preparation of young men for their life work in the profession which is their choice.

They would, however, if that were possible, more fully on their own behalf, and that of their ladies, convey to Sir Donald and to his lady a burden of thankful appreciation for the opportunity of being their guests, which the ordinary words of a vote of thanks are inadequate to express. That they should have had the opportunity to visit his beautiful home, and enjoy his art treasures amid

their distinctive surroundings, is an experience which the members of the society will carry with them to recall with pleasure for many days.

The American Society of Mechanical Enginee s desires to extend to the Montreal Street Railway Company a cordial expression of thanks for the opportunity to visit their power-house during their stay in the city, and for the luncheon served upon that occasion. An inspection of this modern plant, with its present capabilities and future possibilities, was appreciated by all who were able to take part in it.

The American Society of Mechanical Engineers desires to express to Prof. Henry T. Bovey, LL.D., of the Faculty of Applied Science in McGill University, a heartfelt appreciation of the energy and careful thought which has been the foundation of so much of the success of the Montreal meeting.

To Professor and Mrs. Bovey, and to Mr. and Mrs. Frank Redpath, the delighted thanks of the members and their ladies are due for a most enjoyable and successful lawn party, which has been made a feature of their week in Montreal.

The Secretary was further requested and directed, in the name of the Society, to transmit letters of thanks to those who were yet to entertain the Society before the party were to disband finally.

The Society then, on motion, adjourned.

In the afternoon a visit was paid to the works of the Canadian Rubber Company, and later a most enjoyable garden party was given by Mr. and Mrs. J. H. R. Molson at Piedmont Hall. Many representative Canadians were also present.

On Saturday a special train over the line of the Canadian Pacific Railway conveyed the party to Ottawa. Here they were received by a committee, who escorted them in private cars of the Ottawa Electric Railway Company to the saw-mills of J. R. Booth, Esq., where a fine view was had of the Chaudière Falls; to the power-house of the railway driven by water; to the Parliament Buildings, Library, House of Commons and Senate, in which latter room addresses of welcome were delivered by Sir John Thompson the Premier, and by Mr. J. A. Ouimet, Minister of Public Works; thence to luncheon on the grounds of Mr. T. C. Keefer, at Rockliffe, overlooking the river valley; and finally to a garden party held at Government House by Lord Aberdeen, Governor-General of the Province. The water works power-house and the Geological Museum were also open to visitors, but the time of the party was so fully taken up, that this opportunity could not be utilized. The special train conveyed the party back to Montreal, where it disbanded, some members going on by night to Quebec, and others leaving at once for their homes.

DLXXXII.*

TECHNICAL EDUCATION.

BY ECKLEY B. COXE, DRIFTON, PA
(President of the Society.)

On an occasion like the present, when the Society is meeting within the walls of a great institution of learning, it does not seem out of place to devote a few moments to the consideration of a subject which has attracted a great deal of attention in all branches of the engineering profession, and in none more so than the one which this Society represents. I refer to the question of *how shall the young men who are to take our places be trained.*

Within the last quarter of a century, great progress has been made, particularly on this side of the Atlantic, in this branch of education. Schools have sprung up in great numbers, and their standards have been steadily broadening and advancing. Hundreds of young men graduate every year, and, what is more important, find places in all the leading industries. In fact, the command and management of our great industrial army is gradually falling into their hands. Yet those who are directly in charge of their instruction are becoming more and more conscious of the fact that the methods and programmes in use are not wholly satisfactory, and more and more anxious to understand the reason and the remedies. This is shown not only by the devoting of one section of the Engineering Congress at Chicago solely to the discussion of this subject, and by the founding of a society specially devoted to it, but also by the earnest and constant inquiries which are being sent out from many of the institutions engaged in training young men for technical pursuits, asking why their graduates are not as successful in obtaining positions or in filling them as they would wish them to be, and requesting the advice of those with whom they would naturally seek employment as to what changes in the course of study should be adopted in order to render their services of greater value and give them a better chance for advancement. In fact, it is because I have of

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV of the *Transactions*.

late had my attention called so frequently to this matter that I venture to speak to you upon the subject this evening. Of course, it would not be possible in the time at my disposal, even were I able to do so, which I am not, to discuss fully so great and important a question. I can hope at most to make a few suggestions, the results of my business experience, and of what little attention I have given to the subject, which may afford some slight assistance to those who are specially charged with the solution of the problem.

To those who have only looked into the matter superficially, or who are not acquainted with the varied duties which devolve upon the engineering profession, it may seem a comparatively simple question. They may think that the engineering school should be like a school of medicine, a school of theology, or a school of law; but there is a fundamental difference. The doctor, lawyer, or clergyman has practically a well-defined course to pursue, and the professional life-work of a graduate from any of these schools does not differ very greatly from that of his fellow students in the same branch; but we have only to glance through the list of members of any of our engineering societies to realize how widely different are their duties and responsibilities although all may be ranked and known as Mechanical Engineers, or as Civil Engineers, etc.

Engineers are generally classed as mechanical, mining, civil, electrical, hydraulic, military, sanitary, etc.; but there are other classifications which we may consider with profit in this connection. They may be divided into Unsuccessful and Successful. Want of success may be due to many causes, but the most general one (outside of mental and physical weakness) is the fact that the engineer is not adapted to the duties which he is called upon to perform. The old story of the round plug in the square hole! Every one who has had experience in the practical management of business enterprises must have observed that in many cases men who have been comparatively successful have become failures when promoted or transferred to other duties, while in others, men who have previously been only partially successful have become efficient and valuable employees when for some reason they have been obliged to undertake duties different from those in which they have been previously engaged.

The successful engineers, that is, those who have found their work and are doing it well, be they mechanical, civil,

mining, etc., are really either business engineers, constructing engineers managing engineers, theoretical engineers, compilers, or teachers. These terms may not be strictly accurate, but they will enable me, I hope, to express more clearly my ideas of the great variety of the work which is done by the engineering profession. By the words business engineer, I do not mean an engineer who has simply become a business man, but one who understands and can take up any problem, from a strictly technical standpoint, as well as from a business one; and who, after carefully weighing the advantages and disadvantages from both points of view, is able to determine what, under all the circumstances, is best to be done. Such men are very rare, and when once their capacity is known, secure good positions and command high salaries: none are more useful. They know that, while theoretically a certain solution of an engineering problem may be the best, yet, from a business standpoint, another would be much more successful financially. Let us consider for example the mechanical engineer of a railroad. As a simple question of mechanical engineering, the most economical results might be obtained by constructing ten types of locomotives, each of which would do admirably in the condition in which it was to be worked; but as a matter of business, two types constructed as a compromise between these ten might be really much more satisfactory. The cost of construction of each would diminish as the number of types decreased. Repairs could be made more cheaply and easily as spare parts of the two types (which would be interchangeable) could be kept on hand. The greater the number of types, the more difficult and expensive it would be to provide for such interchangeability and the keeping of extra parts on hand. If the business increased in one direction more than another, the compromise engine could still be economical in the new state of affairs; whereas, the specially constructed one would not be for any other work than that for which it was designed. The same remark applies to pumps, cars, machine tools, and hundreds of other things.

Now, I do not wish to be misunderstood in this connection. I am not referring to the engineer who is simply a business man, but to engineers who understand thoroughly engineering problems and can take them up as engineers should in a general way, and bring to bear upon them a business mind and a business training. It is this class of engineers who generally have to determine the

lines along which the next class, that is the constructing engineers, must work. It is their peculiar province, when given the conditions of any problem, to design in all its details the machine, furnace, rolling mill, railroad, or whatever it may happen to be, to decide what is best to be done within the lines laid out, and to see that the work is carried out as decided on. This requires a high order of ability, and in this class we would place some of the greatest engineers the world has known. When once the design has been made, it becomes the duty of the managing engineer to superintend the actual building of the machine, or erecting of the plant according to the plans of the constructing engineer, or to manage it after it is constructed and in practical use. He comes in contact with the foremen and workmen, and must understand and be able to cope with all the problems arising from the management of men. In an establishment where machinery is built on a large scale, his special province would be, after the plans and specifications had been decided on, to obtain proper material and to work up the same; he must utilize to the best advantage the men and machinery, and must understand the relative cost of doing the work in one way or another, and the effect of using high or low priced labor or material; in other words, his engineering training and ability is utilized to produce the best results with the least expenditure of money. This is an absolutely different problem from that of designing or laying out the work. To be successful in this requires a knowledge of men and a broad view of economy; and, in many cases, the success or failure of an establishment is due more to the ability or lack of it in the engineer to whom is confided this part of the business, than to anything else.

There may be subdivisions of each of these three types, or there may be combinations of two or more or of parts of two or more of them in the same man as the varying conditions of the enterprise may require, but the business work of the engineer practically falls into these three categories.

The other three types of engineers whose work is not always appreciated, but the value of which can hardly be over-estimated, viz., the theoretical engineer, the compiling engineer, and the teaching engineer, require a mind of a different kind and a different training. In many cases the same person may be eminently successful in all of these three branches. Under the head of theoretical engineers (words which do not express accurately

what I mean, but which I use for the want of better), I include those who devote their lives to working out the problems which lie at the basis of all good engineering. Such are the men who, familiar with physics, mechanics, chemistry, etc., often without any hope of pecuniary reward, devote months and years to the careful study of the properties of materials and the manner in which they act under different circumstances, and discuss the results mathematically, deducing laws and formulas which become the guide-posts for those engaged in practical work. Of such a character was the work of Sir Lowthian Bell upon the blast furnace; the work of Regneault upon steam, liquids, elastic fluids, expansion of mercury, etc., which laid the foundation for the scientific study of the steam engine; and the work of Weisbach upon the flow of liquids. I have cited only a very few of the great names, but in every country and in almost every locality, you will find men engaged in this class of investigation, the results of which are absorbed in the general literature of the profession, while the names of those who have done it are forgotten by all but a very few. This requires men of special training and special ability. A man peculiarly fitted for it turns to it even when other fields of greater profit are open to him. The testing departments which have been established of late years by some of the large railroads and manufacturing establishments indicate how this branch of engineering has come to be recognized as valuable and to be managed on a commercial basis.

The compiler is another useful but not always thoroughly appreciated member of the profession. It is he who gathers data, which are so widely dispersed in technical literature that it is almost impossible to find them or so badly set forth as to be of little use, and arranges them in such a way that they are easily accessible to the profession or to the parties by whom he is employed. He is often able to obtain from private sources information which supplements and adds value to that which has already appeared; and in many cases by calling the attention of a theoretical engineer to data which are wanting, he is able to obtain them.

Last of all we have the teaching engineer, or professor, by whom the work of those practically engaged in the pursuit of their professions as well as the information obtained by the theoretical and compiling engineer is brought together, digested and transmitted to the students who are being trained to take the

places of those now engaged in the management of the world's affairs. There is no branch of the profession which imposes higher duties and greater responsibilities, or which furnishes greater opportunities for good work. To men who realize these duties, responsibilities, and opportunities, and who are in love with their profession, there is probably no calling which gives more real mental satisfaction; but where the work is not one of love, where the professor is really not interested in what he has to do and does it merely for the purpose of earning his living, there is very little likelihood of his being very successful or contented with his lot.

There is a third method by which engineers who live by their profession may be classified, which though at first sight it may seem peculiar is of vital importance in looking the problem in the face. Engineers may be divided into two classes, viz., those who rise more or less rapidly, and those who do not rise at all, or at least rise very slowly. The engineering profession has been changing for a number of years, and in consequence of this, the condition of the younger members of the profession has changed also. In old times the chief engineer of a railroad did practically everything. He built the road-bed and the bridges, made the frogs and switches, looked after the locomotives and cars, and in fact, was supposed to be able to construct and repair almost everything used upon the railroad. To-day it is different, for the chief engineer of a railroad simply decides that the company will use certain types of bridges, and buys them as he would buy a sewing machine; they are furnished by parties who make a specialty of building bridges. The switches, frogs, etc., with a complete system of interlocking apparatus, are purchased from firms who make a specialty of building them, as are also the locomotives and cars. In fact, the actual work of the chief engineer of a railroad in full operation is often almost reduced to knowing, first, what are the best articles to use and how to obtain them, and, secondly, how to put them together, and use them after you have got them. In consequence of the consolidation of railroads, by which thousands of miles of road come under one head, one man can and does decide such questions for a large portion of the United States just as well as for a road one hundred miles long. The result is that there has been a very great diminution in the number of engineers employed per mile of road, and a great increase in the number of assistants compared

to that of chief engineers, and that there is a large number of engineers in subordinate positions, such as chainmen, rodmen, draughtsmen, etc., for whom there may be little or no chance of promotion; and there are many who must occupy practically the same positions all their lives, as there are very few men above them to whose positions they can hope to succeed. The introduction of the blue-print process has done away with a large number of draughtsmen, and has changed very materially the methods of doing engineering work. A dozen blue prints are now made where two tracings would have sufficed; and as the necessity of comparing tracings after they are made is done away with (the finished drawings being usually made on tracing cloth instead of on paper as formerly), the number and character of the men employed in the draughting office have changed.

If a student after graduating obtains a subordinate position of the kind which twenty-five years ago was the most common (rodman, draughtsman, etc.), his promotion is likely to be very slow, particularly if it is in an old institution, unless he is able to make a place for himself outside of the regular line of his work; but the great problem for the student who has entered upon practical work is how to avoid pushing himself so fast as to hurt his chances, without, on the other hand, standing still. The real truth of the matter is, that in proportion to the work done, the number of men who are engaged in the actual designing, planning and superintending is very small compared with the number of those who are doing more or less mechanical work; that is to say, that the chief engineer with two or three assistants will employ, perhaps, thirty or forty draughtsmen, transit-men, and others of this character, while the amount of work turned out, due to the fact that each man is employed on one specialty, is greater than under the old system. It is nothing more or less than the extension to engineering of the methods of the division of labor. On the other hand, there is a very decided and growing tendency to place the work, which was formerly in charge of men who had risen from the ranks and had become foremen, in the hands of graduates of technical schools, who, after leaving college, have learned the practical part of the business in the works themselves, and, being thus fitted both theoretically and practically for the work in hand, are able in most cases to do better. In addition, there is a desire on the part of many of the more intelligent employers to encourage their workmen who are to occupy places of responsibility to fit

themselves by studying drawing and the elements of mechanics, physics, and mathematics, to better understand and carry out the instruction of those who have the designing and supervision of the work. Not only is this the case, but in Pennsylvania our mining law positively forbids the placing of any mine under the charge of a foreman who has not passed an examination showing that he has, not only the practical experience, but such knowledge of the fundamental principles of mining, with special reference to the subject of ventilation, and of the mine law of the State, as will enable him to conduct the operations under his charge with proper regard for the safety of the men. The examinations are held yearly by a commission appointed by the Court, and those who pass the examination receive certificates of competency from the State, without which they cannot assume the charge of a mine as mine foreman. The result of this has been to force those who wish for advancement to study, and to give us a much better trained set of mine foremen.

The position which the successful engineer occupies to-day does not depend upon whether he graduated as a civil, mining, mechanical or electrical engineer, but upon the fact that he has a mind, and has had a training, which has enabled him to develop in the particular direction in which there was the greatest chance of advancement, in the business in which he was engaged. In other words, you will find mining engineers coming to the front in railroads, mechanical engineers coming to the front in mining, civil engineers coming to the front in metallurgy, etc. There is no better example of this than the late Alexander L. Holley, one of our founders and vice-presidents. When he entered upon his career, there was no branch of engineering in which it seemed less likely that he would make the great success of his life than in metallurgy of steel; and yet there is no one who contributed more to the development of that industry in the United States than he.

Keeping these facts in mind, let us now turn to the question of what is the best way of educating a young man for the engineering profession. If I were obliged to give a specific answer, I could only say, that *I do not know.* Engineering education on the large scale, which our industries now demand, is of so recent a date that we have comparatively little experience to guide us, and in this the other professions have the advantage of us. The problem seems to be: *Not knowing exactly what you want to do, nor the*

material you have to do it with, what is the best way of doing it? In other words, as we do not know exactly what it is best for an engineer to know to make him most useful, nor the exact mental and physical condition of the students who are to be instructed, it is very difficult to determine how to train them to the best advantage. There are, however, certain suggestions which I will venture to offer, not with any idea that they contain much of value, or that they cover much ground, but simply as a contribution to the study of the question.

The subject of technical education has always interested me deeply. As an employer I have come in contact with many engineers, and as trustee of one of our technical colleges I have been obliged to consider it very carefully. It seems to me, in the first place, that the students, no matter what branch of engineering they are to take up, should be trained thoroughly in the elements of mathematics, physics, chemistry, mechanics, and in drawing; and under the head of drawing, I do not mean simply making a handsome drawing, but I mean in the use of the brain, as well as of the hand, so as to understand thoroughly the relative importance of the different elements which go to make up a good drawing. What may be a good drawing for one purpose is a bad one for another, and any expenditure of time or money upon a drawing without increasing its value is to be avoided.

The essential point in training a student, say, in mathematics, is that he should be taught correctly and thoroughly those branches which it is pretended to teach him. In other words, if you do not take him any further than arithmetic, let him be taught properly and understand thoroughly arithmetic; if algebra as far as quadratic equations, let his instruction be real to that point. If he takes up the differential calculus, he should be taught it as far as he goes thoroughly, so that he may understand it. The same may be said of physics, chemistry, and mechanics. If you only teach a man a very little, let that little be taught properly, and let the student when he graduates have thoroughly at his command all those branches which his diploma certifies that he is proficient in. There seems to be an idea in many schools that every engineer should be taken through the differential calculus, and that no one should be allowed to graduate without having mastered it; in fact, in many colleges, as a friend of mine expressed it, the differential calculus serves as a sort of screen which separates the wheat from the chaff; that is

to say, those who pass satisfactorily in the calculus at a certain time, say at the end of the sophomore year, are allowed to go on and graduate, and those who do not are turned back. No one has more respect for the calculus than I have. In my student days, and up to the present time, it has been and is one of my pleasures to study and use it; but my experience in life has proved to me that many men who are not able to really understand it, much less to use it, have been and are engineers of the highest ability, ornaments to their profession, and better fitted for certain lines of engineering work than others of much greater mathematical ability. On the other hand, I have known men to whom the calculus was an easy study, who went through it almost without work, and enjoyed it, and yet these same men were deficient in other qualities of mind, which were absolutely essential for making a successful engineer. I do not wish to be understood as in any way ignoring the great value of the calculus. It has opened fields to engineering science which, without it, it would have been almost impossible to enter; but what I do maintain is this, that there are men who can become great engineers without it, and there are many men who have wasted time in college studying it and trying to apply it without any practical results, and who have never looked at it or used it after they have graduated. As I said before, these are not always the unsuccessful ones, but in many cases the most successful. It seems to me that the calculus should be taught, and taught thoroughly, in technical schools, to all those who are able to grasp it, but that those who cannot should not be prevented from getting an engineering education, as far as it can be given without the use of the calculus, because of their inability to understand it. All good mathematicians are not good engineers, nor are all good engineers good mathematicians.

There seems, also, to be an opinion among a portion of the community that practical training at a technical school, in the machine shop and in the field, is of great importance; in fact, of greater importance than the study of the fundamental principles of the profession. There is no doubt that the practical training of the hand, the eye, and the practical study of the actual operations of machinery, boilers, etc., is of great value; but it is a waste of time to try to make the student a good machinist or a good transit man. He should be taught the practical processes by which the machine work is done, such as

filing, lathe-work, planer-work, fitting-up, etc.; but the time should not be taken from his studies to make him a good workman. He should not spend days in trying to acquire enough experience in surveying to be able to take charge of an engineer corps, for no one would give him such a position. When he graduates, he might get the place of a rodman. If, before or after he graduates, he has time to go into a machine shop, or to go into the field, and do practical work, it will, of course, be of great value to him; but the point I make is, that everything should be sacrificed to training the student thoroughly and fundamentally in those branches of natural science upon which the engineering profession stands. When that has been done, their special application to civil, mining, or mechanical engineering can be given him as far as the time allows; but to sacrifice the former, in order to give him a certain facility in the practical work of any special profession, is a waste of time. These things should be learned afterward in actual practice. In other words, when a man receives a mechanical engineer's diploma from a first-class technical college, it should, and does, mean only that he has pursued with diligence and profit the branches of natural science which the experience of the men who have carefully studied the subject has shown are most useful to those who are to engage in the profession of mechanical engineering. Of course, with it he has been given instruction in the principles of mechanical engineering, and has been directed and allowed to study in practice, as far as possible, their application; but this has been only in the most general way, as it would be utterly impossible for any one in the time that a student passes at college thoroughly to master the details of mechanical engineering. It takes men years thoroughly to study the construction of the locomotive, or of pumps, or of the steam engine. If I wished to employ a young man in an engineering position, and had my choice between two equal in ability, age and health, of whom one thoroughly understood the fundamental principles of mathematics, mechanics, physics, chemistry and drawing, and the other was not as thoroughly trained in these but had a college-shop experience and had gone through the regular courses upon the construction of locomotives, pumps, etc., and was able to discuss more or less intelligently all these different kinds of machinery, I should take the first. Although for the first year or so he might not understand as well as the other the details of the work he was engaged in, and might

require more explanations and go slower, yet, at the end of a couple of years, he would be far ahead; and he would make an equally useful man in any of the branches of the engineering profession; that is to say, he would, as he gained experience, succeed as a civil, mining, or mechanical engineer, because the foundation of his technical training would be so broad and strong that any structure which was constructed upon it would be durable and solid.

A second point which seems to me of importance is the business training of an engineer. He should have some knowledge of business methods and of book-keeping; he should understand the importance of knowing the cost of a thing and the causes of such cost and the methods by which it can be determined; he should realize that no expenditure, however valuable the results which may be obtained from it, should be made unless the money for it can be provided without crippling other branches of the business; and he should know that it is not good engineering, that it is not good business, to let a contract at a price which he feels sure is below the actual cost of doing the work, for, except in rare cases, one of two results will follow: either the work will be slighted, or the contract will be abandoned, both of which may occasion a great loss. I cannot be too emphatic in regard to this. It is hard, I think, for any one who has not had actual experience in the matter, to realize how much the usefulness of an engineer, even when his functions are not of great importance, is lessened by want of business training and a knowledge of business methods. Many a young man who had a promising career before him has been prevented from advancing because his employer did not feel safe in confiding anything of a business nature to his charge.

The engineering student should be taught the absolute necessity of accuracy of observation, accuracy of statement, and accuracy of reasoning. He should be particularly cautious if superficial observation seems to show that the results are as hoped for. When things do not turn out as we wish, we are very likely to examine every detail with the greatest care; but when they seem to be as we desire, we are not always so particular. When an engineer has the slightest doubt as to the result which he has obtained, or the conclusion at which he has arrived, he should state these doubts with great fulness and as specifically as possible when making a report. It may not please your client at first, but it will please him much more than if you

leave them out and he should find afterward that, had they been taken into account, failure might have been prevented. Above all, he should be taught not to waste. The greatest claim that the engineering profession has upon the respect of the world is that it has persistently and continuously worked to save waste, and to convert those things which are hurtful or useless into sources of wealth.

The technical school should be like any other great engineering establishment. It should aim to obtain with the material at its command a product which should have the greatest total value. It should not try to turn out a few very brilliant men, while the great majority of the students are either dropped or imperfectly educated. Of course a school, like the blast furnace or rolling mill, should endeavor to get the best raw material which can be had in sufficient quantities and at a reasonable price, and it should furnish a product of the greatest value in the markets to which it has access.

Men who become distinguished as engineers, lawyers, physicians, or clergymen may do so because they possess one special faculty which is particularly valuable in their chosen profession, and which places them in some way above the mass of their colleagues. This type of man, if he has the misfortune not to adopt the profession for which his talent peculiarly fits him, is very unlikely to make a success. Other men become distinguished because they have good, broad, all-around minds which enable them to take up and master any profession which they may adopt. A man of this kind of mind is likely to succeed no matter what calling he may select and no matter under what circumstances. It is rare for a man who has made a success in one business or profession to make an equally great success in another, because the successful men are generally men of the first type; but when a man has a mind of the second type, he is likely to succeed in any business which he undertakes, and sometimes he may make a greater success in the second than in the first.

It seems to me, therefore—but I express my opinion with a great deal of diffidence—that the technical school should strive to produce rather well-trained, all-around men, than specialists; that is, that the training in those branches which form the groundwork of all engineering should be uniformly thorough, none being carried to a higher point than the others and none being neglected. In other words, that the student should have as much

knowledge of each of the branches as is necessary for him to thoroughly utilize his knowledge of the others.

The students, under these circumstances, could enter almost any establishment and very soon make themselves very useful men by further training, instead of being specialists, who, if they should not obtain employment in their specialty, would be practically of no more value than untrained men.

The faculty of a technical school should also consider very carefully the condition of life from which its students are mostly drawn, and the occupations, or positions, which their students will be most likely to fill after they graduate.

DLXXXIII.*

A NEW FORM OF CANAL WASTE-WEIR.

BY JOHN R. FREEMAN, BOSTON, MASS.
(Member of the Society.)

THE ordinary type of canal waste-weir is a simple straight dam of masonry or timber with an open rollway. At Holyoke, Lowell, Lawrence, Lewiston, Manchester, and at many other water powers, the same general model is followed with but little variation.

Three years ago the writer was called upon to design a suitable waste-weir for the canal of the Nashua Manufacturing Company at Nashua, N. H., and although there was nothing in the circumstances to have prevented following the old familiar model, a closer study of the problem resulted in devising a type of structure which has since been found so successful in its installation and operation, and was withal so economical to construct for a given length of overfall, that a brief description may be of interest to some of the many members of our Society who have to deal with water power.

The canal in question is about two and a half miles in length, and conducts the whole ordinary flow of the Nashua River to turbines of the Nashua Manufacturing Company's cotton factory. This flow within the canal, during working hours, is ordinarily about 600 cubic feet per second, which develops about 1,800 H. P. If all the present turbines were in operation at full gate, about 2,800 H. P. would be developed. The canal for nearly its entire length winds along the river bluff, which is of sand and gravel and is 20 to 30 feet high. The embankment which restrains the water is pure loose sand, without masonry wall, puddle core, sheet piling, or clay, but which has become tight and impervious from the silt and consolidation during more than forty years of use.

The top of the embankment is in many places but little more

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

than two feet above the high water line, and the head gates, by which the height of the water in this canal is controlled, are more than two miles distant from the factory by the shortest road.

It is, therefore, of great importance to have an overflow weir of large capacity which, like a safety valve, shall prevent the water from ever rising so high in the canal as to endanger this long and easily destructible embankment.

It is also of great advantage to have a waste-weir and sluice-way of such compact form that it can be placed near the factory buildings, in plain sight of every passer-by and quick of access, and to have the waste-way channel obstruct the factory yard to the least possible degree.

One further requirement was that this waste-way should afford the utmost convenience for getting rid of ice in the canal, which, in the severe winters of this region, forms over the canal freely during the night, and is broken off at times by the rapid current of the day and comes down sometimes as a great quantity of thin sheets, or, at other times, becomes detached in large, thick cakes.

Such requirements as just mentioned are found at many of our Northern water-power mills.

The form of apparatus herein described has fulfilled all the requirements of the situation, in a manner giving satisfaction to all concerned, and is presented to the attention of those members of the Society interested in water power, in the belief that there are many situations where this general hopper-like form of plate iron, with the discharge carried off with the utmost safety to the embankment through a large plate iron tube, will be found advantageous.

The general surroundings are shown in Fig. 190. The embankment *A*, *B*, *C* is an embankment of pure loose sand, and probably is at least 40 feet deep at *B* above the underlying ledge. Evidently, should any rupture of the bank occur, the high and very valuable factory buildings between it and the river would be imperilled. Most of the embankment *B D* is forty or fifty years old, and, therefore, although of pure sand, has become well compacted.

No continuous sheet piling had in earlier years been carried along through the embankment *B D*, and we added none now, except the lines at *A*, *B*, and *C* (Fig. 192), each line only about

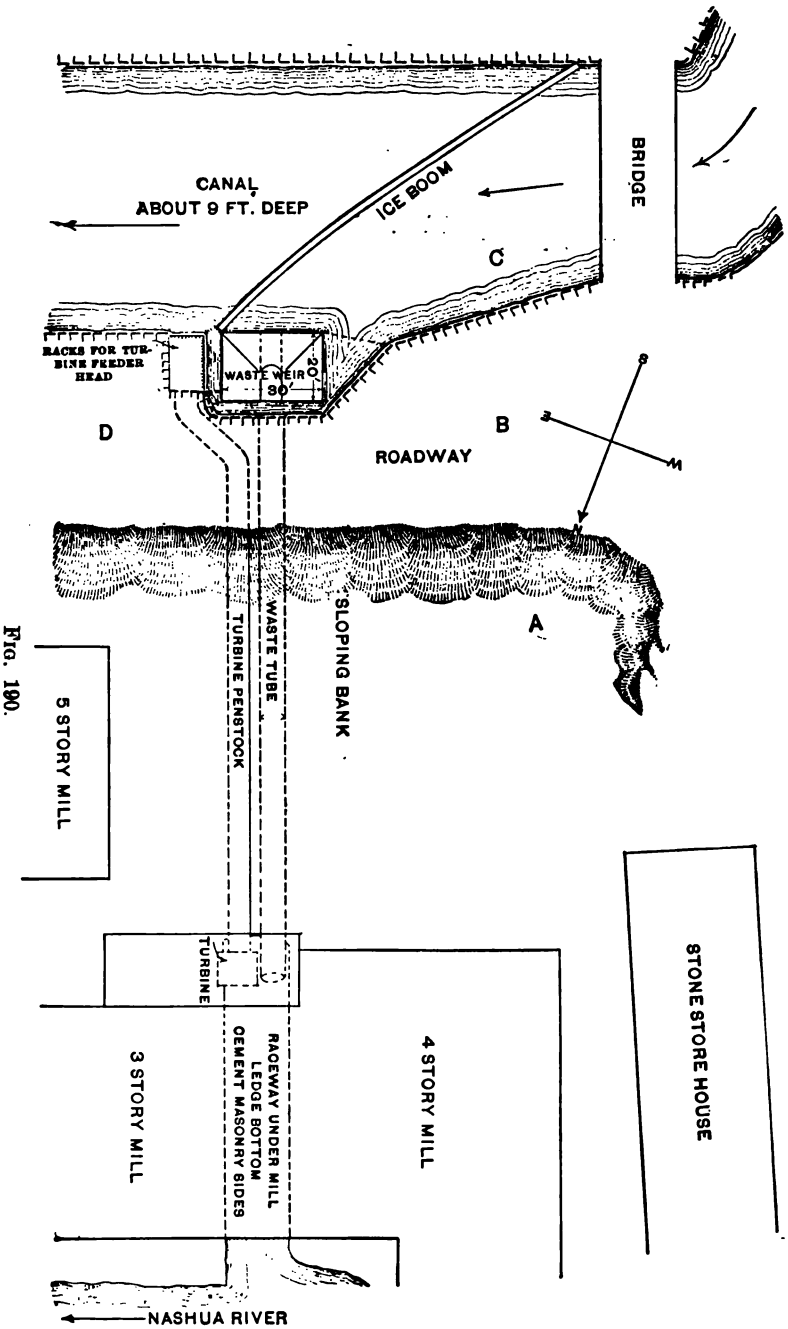

BRIDGE
ICE BOOM
CANAL
ABOUT 9 FT. DEEP
C
RACKS FOR TURBINE FEEDER HEAD
WASTE WEIR
20'
30'
D
B
ROADWAY
A
SLOPING BANK
WASTE TUBE
TURBINE PENSTOCK
5 STORY MILL
TURBINE
RACEWAY UNDER MILL
LEDGE BOTTOM
CEMENT MASONRY SIDES
3 STORY MILL
4 STORY MILL
STONE STORE HOUSE
NASHUA RIVER

FIG. 190.

40 feet long, but in order to prevent water from leaking and following down along underneath the penstock, cut-off walls of masonry were built around the lower half of the penstocks, 15 feet apart, each 2 feet thick, and extending out from the iron conduit a distance of 2 feet below it and from 2 to 4 feet out from its sides.

The waste-weir is formed along the edge of a hollow rectangle, 20 feet × 30 feet, which is surrounded by water on all four sides, and it thus presents a total length of overflow crest along the flashboards of nearly 100 feet.

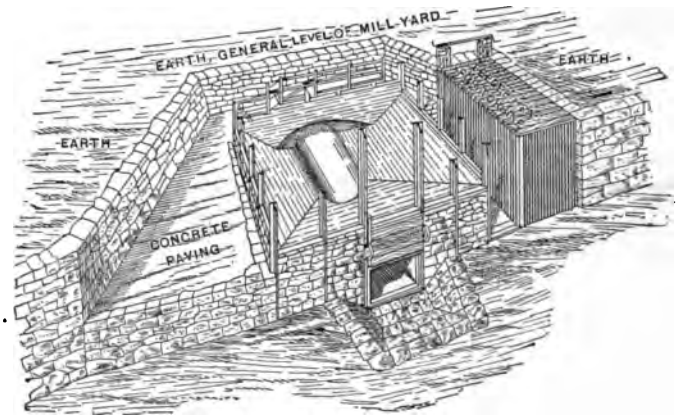


FIG. 191.—Sketch showing form of Weir—Canal empty. Flashboards and Hoisting Rigging Platform around top of Weir not shown.

This great length of overfall is thereby obtained in a very compact form, and in a space so small that it could easily be roofed over, or covered by a house if special conditions required this.

It was at first the intention to build a house about 38 feet × 28 feet over this weir, which should protect it from the extreme cold, and thereby prevent ice building up on its interior, as water leaked through the flashboards, or slowly ran over them, and which should also form a shelter for any men who might have to be kept working on the weir when the ice was running badly in the canal. It was concluded to defer the building of this house until its absolute need should be shown.

The winter following the completion of the structure was exceptionally severe, but everything went along all right without any house, and during the winter just passed it was also

found that no house was needed over the weir. Therefore none will be built.

The declivity was so steep, that the ice, from small streams running over the weir, did not attach itself to the smooth steel plates of the interior, nor did it catch on within the waste-way tube. No interior projection existed to retard the water and give the ice a chance to form; the rivets were countersunk, and the iron plates were made up with smooth butt-joints on the inside of the hopper with butt-straps beneath.

Moreover, it was found that this ice-way could pretty nearly look out for itself. The openings between the flashboard handles were so ample; the "piers," or vertical deck beams, presented so narrow an obstruction; and more than all, the "gate-flashboard," at the middle of the front, presented so generous an ice-way when hoisted, that no laborers were needed to be stationed on the weir when ice was running.

Fig. 192 is a longitudinal section corresponding to Fig. 190, and illustrates very clearly the small amount of masonry required for the waste-way proper, and the safety against any possible scour from the water after it has passed the crest of the weir.

This particular structure was complicated by the large drain gate and the removable flashboards, but it has appeared to me that a simple design like Fig. 193 might sometimes prove a useful means of providing an overflow through an earthen embankment, taking, of course, suitable precautions to strengthen the waste-pipe between *E* and *F*, if the bank were high, or, if there were great values at stake, by

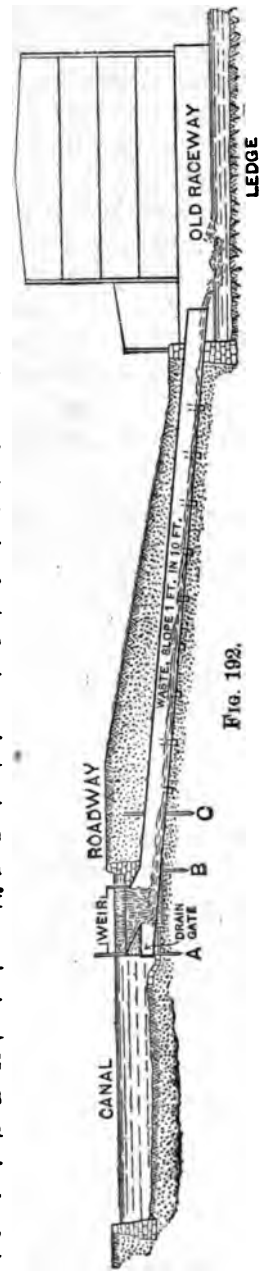


Fig. 192.

encasing it within a mass of concrete resting upon natural hardpan, or resting upon other suitable foundation. The waste penstock should of course be provided with suitable cut-off flanges or walls, as *G*. The bottom and sides of the hopper should slope steeply, and enter the tube with no corners or angles. A very high declivity should be given to the tube itself, so the throat at *E* cannot by any possibility become gorged, and the bottom of the outlet *J* should be above the tail-water, lest it become partially obstructed by ice forming within its lower end.

In the particular structure described, the bottom of the hopper was in smooth alignment with the tube. The side slopes of the hopper merged into the curve of the invert of the tube with no abrupt angle, and in a form which experiment on a model had indicated would roll the current of water or blocks of ice back into the channel with a smooth and speedy exit.

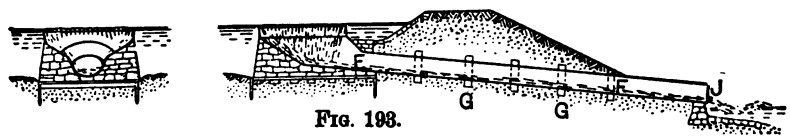


FIG. 193.

Special care was taken to provide a waste tube of such declivity, shape and area, that a gorge could not form at the throat. The declivity of the tube was so great (1 foot in 10 feet), and its area so generous (eight feet in diameter), that it would run but little more than half full (as indicated by a theoretical computation) with every inlet opened wide, all the flashboards pulled out, the drain-gate raised, and the canal at high water mark.

This weir was provided with removable flashboards on all four sides. On the front the iron overflow sill of the permanent structure was 3 feet lower than the ordinary water level, on the two ends the permanent sill was 2 feet lower, and at the back 1 foot lower.

The flashboards were each 1 foot wide; and thus on the front, for instance, the crest could be lowered 1 foot, 2 feet, or 3 feet, according to any temporary need.

Experience on the main water power canal at Lawrence had taught the writer that the best way to run out ice with the minimum waste of water is to let the broken ice accumulate in

the boom above the weir for a time, meanwhile wasting no water, then to open from one to four bays of flashboards 3 feet deep and sweep it out with a rush, and then shut up these flashboard bays, save water, and let the ice accumulate for a while again. Experiments made there show that a current of the force produced by an open gap, 3 feet deep, is needed to sweep the ice along well, and that a 5-foot gap is rather too narrow to pass heavy ice without clogging. This gap, 7½ feet long by 3 feet deep at the Nashua weir, serves admirably, as is illustrated by the photograph reproduced on page 683.

The depression in the water surface close to this large opening cracks the larger cakes of ice, and the strong current and this large depth at the crest quickly sweeps them over out of the way.

This flashboard gate can be lifted its full height in about half a minute, and will run down under its own weight in, say, 5 seconds, and the quickness and ease with which it can be opened and shut adds much to the facility with which one man can manage all when ice is running badly.

The following figures illustrate the amount that might come over the weir with flashboards pulled out, and canal at full height:

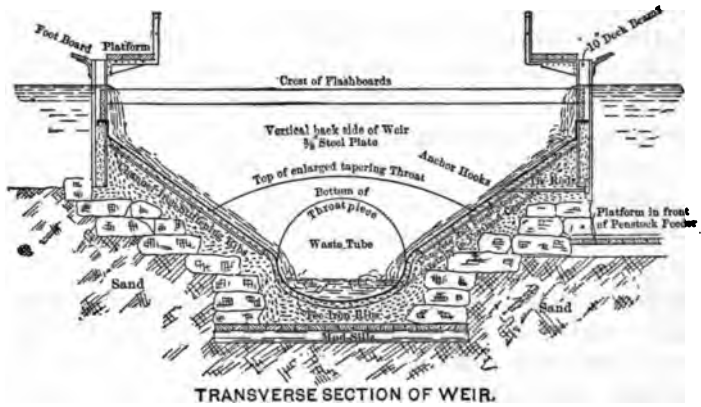
APPROXIMATE DISCHARGE OF OPENINGS IN WEIR.

	Cubic feet per second.	Equivalent horse-power or turbines.
Flashboard gate, 7½ feet × 3 feet, wide open	120	360
All remaining four front flashboards removed, 3 feet deep	300	900
All eight end flashboards removed, 2 feet deep	320	960
All five back flashboards removed, 1 foot deep	90	270
With every flashboard around weir removed	830	2490
Draw-off gate, 7 feet × 4 feet, wide open	500	1500

Therefore, either through the drain-gate alone, or through the flashboard spaces alone, an amount of water, about equivalent to the whole draught of the turbines, could be wasted should any sudden accident demand the shutting down of the turbines without opportunity to close the gates at the head of the canal, 2½ miles distant.

On the other hand, if all flashboards and gates at the weir

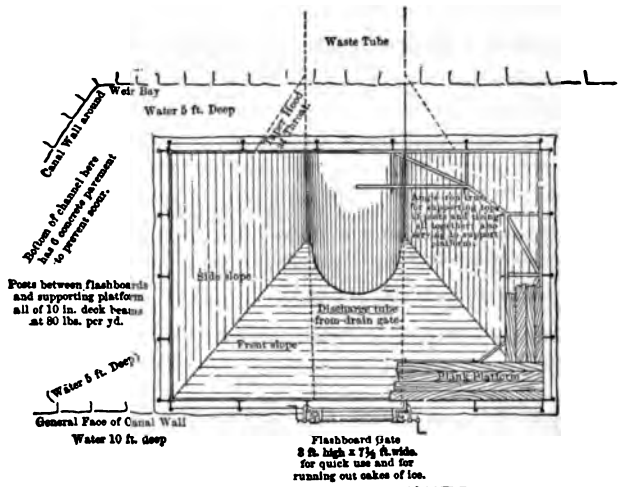
were left shut, when suddenly closing the turbines, while the gates at the head of the canal were left open, a rise of 1 foot in the canal-level would pass 300 cubic feet per second over the crest of the weir, and a rise of about 18 inches would pass the whole ordinary flow of the river over the waste-weir.



TRANSVERSE SECTION OF WEIR.

FIG. 194.

The description of the details may be very brief, since all is shown with tolerable clearness in Figs. 194, 195, and 196. We



GROUND PLAN OF WEIR.

FIG. 195.

may note that all parts between wind and water were of iron or steel, that the rivets were countersunk on all interior surfaces

awash, and that all extended surfaces of steel plate against which might come the impact of cakes of ice and falling water, were so backed up by concrete as to give them an anvil-like solidity, and that to prevent the ⅜-inch steel plate from springing away from its masonry backing, all extended surfaces were stiffened by ribs of channel iron which hooked into the masonry, or by anchor bolts at frequent intervals.

Everything about the water-way was made smooth as possible, without projection, and with no edge of a sheet and no rivet head projecting against the current where it could be struck by moving ice or floating material.

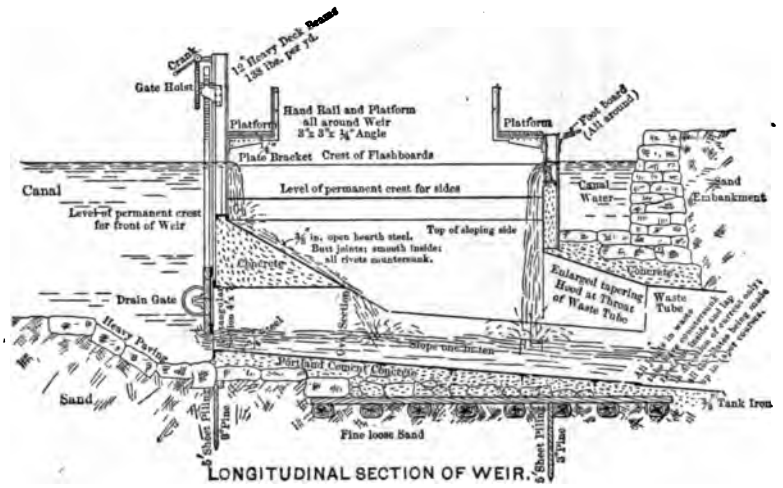


FIG. 196.

The contractor bid almost as cheaply for the waste tube built in tapered courses lapped in the down stream direction, so that the edge of no sheet projected against the current, and with all rivets in the lower half of the section countersunk, as he would if it were built in "large and small" courses and with projecting rivet heads.

The upper part of the cross section was a semicircle, but the lower half of the tube was made of a broad shallow elliptical section (shown in Fig. 194) to facilitate ice being washed through by the water.

The flashboards were designed with handles of a novel form, as shown in Fig. 197 perspective and Fig. 198 in plan.

Two or three-high flashboards as commonly constructed have

the pair of handles by which each board is pulled, bolted to the back side of the board, and with the handles of the top board nearest the centre, those of the next lower board a few inches farther out, and so on. All of these handles shorten the over-fall and seriously obstruct the weir from acting as a safety overflow. With a flashboard 4½ feet long by 1 foot

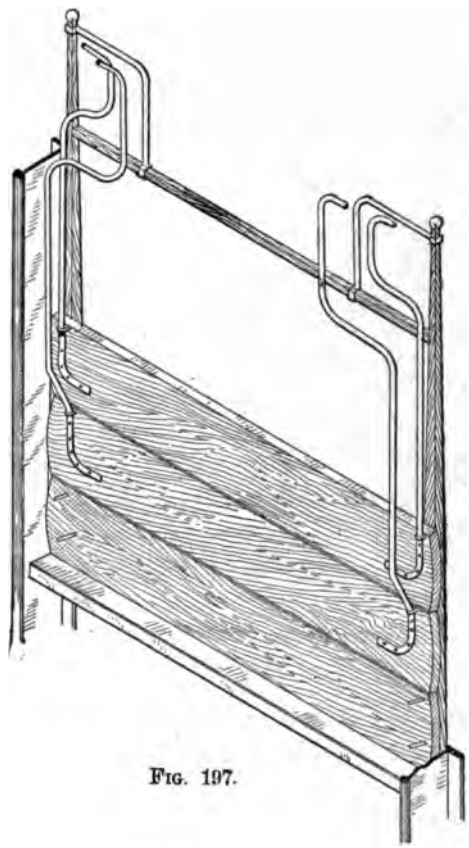


Fig. 197.

deep, which is about as large as one man can pull with a foot or two of water pouring over its top—and with these boards in series 3-high—the six handles of ordinary size would obstruct more than one-third of the whole crest-length, besides so subdividing it that floating twigs and other rubbish find lodgment and cause further obstruction.

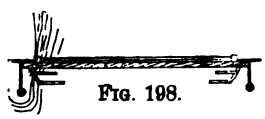


Fig. 198.

The handles of the two uppermost of these flashboards were made of one-inch wrought-iron pipe stiffened by a bar of Norway iron welded into its lower end where it was attached to the board. The handles by which the bottom front boards pulled were of wood and rested congitudinally against the flange of the deck-beam pier. As these were too wide apart for a man to grasp while lifting with his full strength, the cross bar was added as shown to the handles of the lowest board. The tubular iron handles of the two upper boards were bent inward to a convenient distance, such that the weir attendant can exert his full strength in lifting, and at the same time the board can be tipped forward so that the pressure of the water against its inclined under side eases the lift.

The point to be noted with special interest is that all the handles were kept out at the full length of the board so that the length of overfall was not appreciably shortened by them.

In point of fact, the carpenters here happened not to consult the drawings when they reached this stage, but patterned their handles by the old-fashioned model, and in the hurry to get through, we changed over only one set of them to the form designed. Experiments with these show the new form satisfactory.

The foot board shown in Figs. 194 and 196, is a great aid in furnishing good standing ground when a flashboard starts hard, and adds much to the safety of the man.

The steel deck beams were found to present great adaptability for this method of construction, and the two tall, extra heavy 12-inch beams in front gave very advantageous supports for the two sets of hoisting rigging, for the drain gate and the flashboard gate.

In these northern latitudes the writer has found a broad, deep flashboard, geared with rack and pinion, so that one man could easily lift or close it quickly and surely, of great advantage when running out ice and useful if a turbine was temporarily shut off. We therefore provided here a flashboard gate $7\frac{1}{2}$ feet × 3 feet in the clear, of $2\frac{1}{2}$-inch pine, with gate-starts at each end of $3 \times 2 \times \frac{3}{8}$ inch angle iron, to which cast-iron racks of $\frac{7}{8}$-inch pitch and $2\frac{3}{4}$-inch face were bolted. These were lifted by a 12-tooth pinion coupled on same shaft with a 74-tooth gear into which a 15-tooth pinion worked, actuated by a hand-crank

of 15-inch radius. One revolution of the crank lifts the gate about $2\frac{1}{10}$ inches, and this gear ratio is found satisfactory in practice as to power and speed.

The gate for drawing off the canal for repairs was necessarily large, for the canal was long, the volume of water large, and

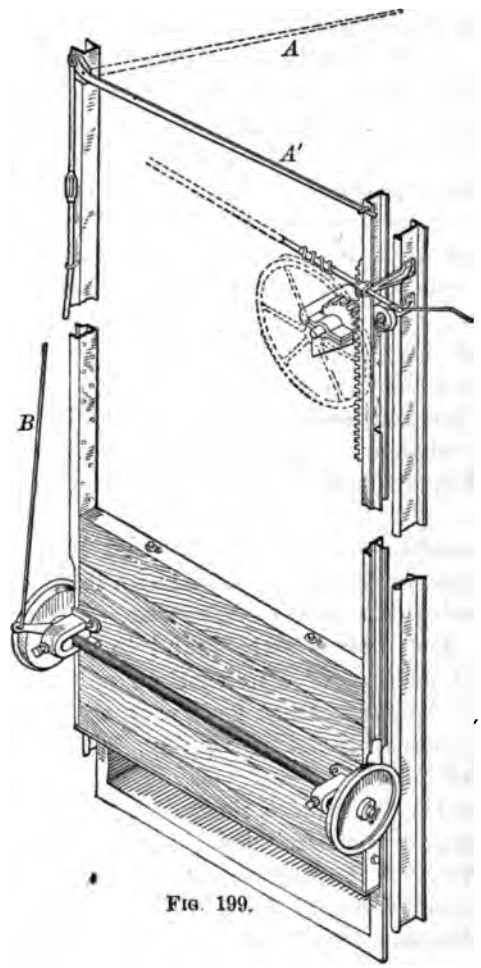


FIG. 199.

promptness of action might be needed. The opening is 4 feet × 7 feet in the clear, and the working head upon the centre of the gate is about 10 feet. The total pressure of the water

against the gates is 10 tons, and the pull required to lift it with 40% friction and including weight of gate would be 5½ tons and possibly more if surfaces were left in contact for a year without motion, rusting together.

In order that this gate may be lifted *quickly* by one man, as a night watchman, it was necessary to avoid the ordinary waste of power by friction, therefore the trucks shown in Fig. 199 were added. The axle bearings of these trucks are bushed with brass to prevent rust.

The lever A stands nearly vertical when the gate is closed. When it is desired to raise the gate this lever is pulled down and held under a hook as shown at A′. Through the rod B the axle of the trucks is turned about ⅙ of a revolution, and as the journals of this axle resting upon the gate are eccentric with the trucks, the trucks are forced in hard against the rails by this turning of the axle, and the main gate is thereby pushed off about ¼ inch from its seat. Sliding friction of the gate against its seat thenceforth ceases.

The action is much as if we loaded our gate on a wheelbarrow instead of crudely dragging it along the surface. One man alone can lift the gate its full height against a full head of water in about five minutes.

In some experiments made a few days ago by Mr. John Whitney, master mechanic of the corporation, it was found that a weight of 33 lbs. hung from one of the hand cranks when in a horizontal position, with a lever arm of 15 inches, would lift the gate when the lever A was pulled down and the trucks in service; but that 145 lbs. weight applied in the same manner was needed to move the gate with the lever A thrown up and the trucks therefore out of commission.

A gate of the same area at the Lawrence water power, and with about the same speed of hoist, when working under 10-foot head (as with a canal nearly empty) requires the united strength of four men to start it, and five or six men sometimes have to work at one crank.

At this Nashua weir gate provision was made for hanging a weight of some 30 or 40 lbs. from the end of this long lever A, which would counterpoise very nearly the pressure of the gate against its seat and therefore continually carry so great a portion of the load upon the trucks as to make hoisting easy with no further attention; but it was found so simple a matter to pull

down this lever by hand and catch its end under a hook as to render such a counterpoise needless.*

Another means for lessening the great waste of power common in structures of this kind was a better form of worm. It is strange to the writer that millwrights will persist in the use of cast-iron worms, generally 7 or 8 inches outside diameter, for $1\frac{1}{2}$-inch pitch, when the use of smaller diameters will effect so great a saving.

We cut these worms from a solid steel forging, and made the diameter the least that theory and experience would sanction. At the same time we thinned the wrought-steel worm tooth one-eighth of an inch, and added this to the cast-iron worm wheel tooth to strengthen it.

The elements of the gearing are

2 main gate-starts—rack teeth 5 inch face, $2\frac{1}{2}$ inch pitch—involute tooth $1\frac{11}{16}$ inches thick at root.

2 main pinions, 10 teeth 6 inch face, shrouded both ends.

2 worm wheels, cast-iron cut teeth, 39.8 inches pitch, diam. 84, teeth 3 inches length (corners clipped).

2 worms cut from forged steel blanks, $3\frac{5}{8}$ inches outside diam., $1\frac{1}{2}$ inches pitch.

Fig. 200 is from a photograph taken in the winter of 1893, showing the operation of the waste-weir and the conditions it has to fulfil in a rigorous climate.

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* Although this hoisting apparatus at Nashua has proved very satisfactory, I consider that for larger gates the eccentric shaft furnishes a less desirable means for lifting the pressure off from the face plates and on to the trucks than can be obtained by wedge-like depressions at the bottom of the rail into which the truck falls when the gate is at the bottom, but which as the gate is laboriously raised the first inch or two, brings the trucks to a firm bearing on the rail and transfers to them the load so that it is lifted the rest of the way with ease. I have made use of this form of device in three more recent designs where the load was to be unusually heavy, and have furthermore provided roller bearings on the trucks and ball bearings on the worm-shaft, with much advantage. Perhaps the most interesting example, though not the latest development, is at the large sluiceways and canal head-gates recently completed from my designs, at Sewall's Falls on the Merrimack River, above Concord, N. H., at which gates 10 feet wide × 12 feet high can be operated by the strength of one man, providing the gearing is properly oiled and cared for.

These Sewall's Falls sluiceways and the apparatus connected with them are described in a paper soon to be published in the *Transactions* of the American Society of Civil Engineers.

FIG. 200.

The cost of this waste-weir was approximately as follows:

Coffer dam in canal.....	$ 700	
Excavation and timber platform	400	
Masonry and concrete....	1,500	
All iron and steel work on hopper and throat of waste-weir penstock....	2,500	
Gates, hoisting, gearing and flashboards	1,000	
Total for waste-weir proper.....about		$5,410
Waste penstock 185 feet long, in place.....	$2,900	
Excavation backfilling and cross walls.....	500	
Total for waste penstock		$3,400
Total for weir, waste-penstock, and coffer dam.....about		$9,500

DLXXXIV.*

A NOTE ON COMPRESSED AIR.

BY FRANK RICHARDS, NEW YORK CITY.
(Member of the Society.)

In offering the following paper for the consideration of The American Society of Mechanical Engineers, which embraces in its membership the ablest of investigators of mechanical efficiencies, it is not intended to challenge minute and exhaustive criticism, but rather to present the subject in such a way that it may attract as widely as possible the attention of the general mechanical public. I may be able to make it appear that the compressed air practice of the day is not sufficiently advanced to invite or to warrant the minute and careful analyses which belong, for instance, to the marine engine, to the pumping engine, or to the locomotive.

I think that compressed air has not received the attention which it deserves, and that where it has secured some attention it has not always had fair play. There are many men in these days, and many intelligent engineers among them, who will not even consider the claims of compressed air as a means of power transmission, because their minds have been so filled with the idea that its use entails enormous losses. That consideration settles it for them, and that is the libel from which compressed air suffers, so that it does not get a fair chance even to show what it can do. It is only another case of giving a dog a bad name; and in this case it is a very good dog with a very bad name. It is the worst kind of a case to set right. It is but common justice to tell the straight truth in the matter and to spread it as widely as possible.

The position of compressed air before the mechanical public, and especially the American mechanical public, has been a peculiar one all the way through. It has had no disinterested,

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* Presented at the Montreal Meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

all-around friends to look after its interests, nor interested ones either. There have been no men, and still less any company of men, who have made the application of compressed air their business and have looked after it. Where is the general compressed air company performing for compressed air such functions as more than one company is performing for electricity? and why is there not such a company? Of the builders of air compressors not one of them has been, not one of them is to this day, responsible for the economical application of compressed air *after the compression*, or apparently cares anything about it. Where compressed air has been used in this country, and where any thought has been given to economy in its use, the air compressor, it would seem, has been almost exclusively studied and talked about. In the progress of the steam-engine it has sometimes seemed that the steam-boiler has hardly received its proper share of attention. In the existing writings upon steam-engine economy it is probably safe to say that the engine engrosses ten times as much of the matter as the boiler does. In the case of compressed air the boiler, or compressor, gets ten times as much study and discussion as the engine or motor which uses the air. There are such vagaries of injustice in civilized communities.

And this impression which has got abroad, of the waste of power in the use of compressed air, has, curiously enough, been promoted and disseminated by the air compressor people more than by anyone else. We may say this with perfect safety, for it strikes so generally that it hits no one in particular. The compressed air literature accessible to the general public consists principally of air compressor catalogues. The argument of the average catalogue runs like this: "If you are going to use compressed air for any purpose, look out for the enormous losses of power to be encountered and which you are sure to experience *if you don't buy our compressor*." And the result has been that many have "caught on" to the terrible tale of the waste of power, and have helped to spread it far and wide. The argument is of course not maliciously meant, but it has done more work and somewhat different work from what was intended.

Fig. 201 tells the whole story of compressed air. The piston F is fitted to the cylinder E so that we may assume it to move freely and without leakage. The piston being at A, as shown,

and the cylinder being filled with air at a pressure of one atmosphere, and at normal temperature, a sufficient weight is placed upon the piston to force it down into the cylinder and compress the air contained in it to a pressure of say six atmospheres. The volume being inversely as the pressure, the piston should go down to *C*. We find that it actually only goes down to *B*, and the reason is that while the air is being compressed the operation of compression also heats it, and the heat distends or expands the air, and its volume is consequently considerably greater than it should be upon the assumption that the volume is always inversely as the pressure. Supposing both the piston and the cylinder to be absolute non-conductors of heat, and that the air heated by the compression loses none of its heat of compression, then if the weight which forced the piston down be taken away the piston will be driven back to its original position at *A*, and the air contained in the cylinder will have resumed its normal pressure and temperature, and will have done as much work, or will have exerted as much force, by its return as was employed in the act of compression. If while the piston was at *B*, and with the weight upon it sufficient to balance the pressure of six atmospheres, the air by any means had been cooled to its original temperature, the piston would have fallen to *C*, and the law that the volume varies inversely as the pressure would have held good, as then the initial and the final temperatures would have been the same. The air being thus cooled to its original temperature, and the piston being at *C*, upon removing the weight from the piston it would return only to *D* instead of to *A*. When the piston arrived at *D* the pressure of the air in the cylinder would have fallen to the original pressure of one atmosphere, and the piston at *D* would be balanced between the equal pressures above and below it. As the air is heated during the operation of compression, so is it equally cooled during its expansion, and when the piston reaches *D* the air in the cylinder is then at atmospheric pressure, because it is then much colder than it was at the beginning; and it is

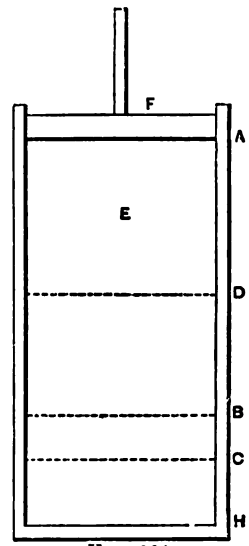


FIG. 201.

solely because of this loss of heat that the pressure falls so early, and that the piston does not return to A where it started from. If while the piston is at D the air can by any means recover all the heat which it has lost, the piston will return to A as before. The distance DA compared with CA represents the total possible theoretical loss of power in the compression and the re-expansion of air.

We may now consider the practical aspect of the case. And, beginning at the very beginning, with the practical compression of the air, we cannot but notice how readily the air lends itself to economy in its use. By the way in which it is possible to connect the air-compressing cylinder with the steam-engine in the best air compressors of the present day, no power is lost in operating the piston of the compressing cylinder; or, stated more accurately, the entire friction of the air-compressing apparatus is so compensated for by the saving of friction in the engine that it is not only reduced to zero but it becomes a negative quantity of considerable magnitude. This may be proved by incontrovertible evidence. The air compressor, it will be noted, comprises a complete steam engine, with all its component operating parts, steam piston, crosshead, connecting-rod, crank-pin, crank-shaft, complete valve motion, etc., and in addition to these there is the air piston to operate, and it is natural enough to suppose that the operating of the air piston puts just so much additional friction upon the engine, with a call for so much more power to overcome it. But the arrangement of the air piston in a straight line and upon the same rod with the steam piston saves enough from the friction of the engine proper to more than compensate for the friction of the air piston, and the operating of the air piston costs practically "less than nothing." In a regular stationary engine which transmits its whole power through the crank shaft, and by its driving pulley, gears, or otherwise, wherever it may be wanted, the whole power exerted is felt in friction upon the crosshead slides, connecting-rod boxes, crank-pin, main bearings, etc. In the air compressor most of the power of the steam-engine is transmitted directly from the steam piston to the air piston without the intervention of the engine friction. Not only is the effective steam pressure so immediately transmitted, but the force due to the inertia of the reciprocating parts, especially the heavy crosshead which is usually employed, is sent directly to the air piston through its piston-rod.

The losses due to the friction of the ordinary stationary steam-engine usually amount to 10 or 12% of the indicated horse power. In an Ingersoll-Sergeant air compressor exhibited at Chicago, with engines of the Corliss type, the mean friction, as determined by Professor Jacobus, was 5%, so that, crediting the saving of friction to the air cylinders, the friction cost for operating them was −5 to −7%.

The diagram, Fig. 202, scale 40, is intended to show the practical possibilities in the use of compressed air at 75 lbs. gauge, or six atmospheres. The line *a b* is the adiabatic compression line, or the line of compression, assuming that no heat is taken away from or lost by the air during the compression. The initial temperature of the air being 60°, the final temperature would be

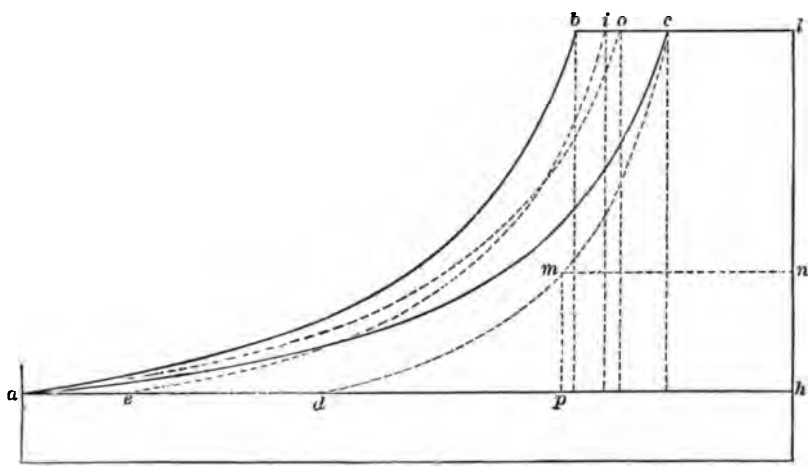


FIG. 202.

about 415°, and the final volume is .28 of the initial volume. The line *a c* is the isothermal compression line, which assumes that all the heat of compression is got rid of just when it is produced, or that the air throughout the compression remains constantly at its initial temperature. The final volume in this case is .1666 of the initial volume. Remembering that these lines, *a b* and *a c*, represent the compression of the same initial volume of air, it is evident that there is quite a difference in the power employed in the two cases, and herein lies the loss, or the possibility of loss, of power in the operation of compression. The mean effect-

ive pressure, or resistance, of the air for the stroke on the adiabatic line *a b l* is 35.36, while the mean effective pressure for the isothermal line *a c l* is but 27, or only 76% of the former. The comparison should, however, be made the other way. The adiabatic mean effective pressure is 130% of the isothermal mean effective pressure, and the 30% is, of course, the additional, or, as we might say, unnecessary, power employed. Neither of these lines, *a b* or *a c*, is possible in practice. Air cannot be compressed without losing some of its heat during compression, so that the actual compression line must always fall within the line *a b*. On the other hand, it is equally impossible to abstract all the heat from the air coincidently with the production of that heat, so that the actual compression line must always fall outside the line *a c*. The best air compressor practice of to-day is very near the line *a o*, or the mean of the adiabatic and the isothermal lines. The actual line is generally outside of this, and seldom inside of it. It would be impossible to produce a line exactly coincident with this in practice. If we produced a line giving the same mean effective pressure as *a o*, it would probably run above *a o* for the first half of the stroke and perhaps a little below it at the last. If the compression were "compound," or done in two cylinders, there would, of course, be a break in the continuity of the curve. The mean effective pressure for the line *a o l* is about 31.5, or still nearly 17% in excess of the mean effective pressure for the line *a c l*. As this is for what must be considered exceptionally good practice, the loss of power for the average practice in air compression may be put at 20%. Lest some impatient ones should drop the subject here, or lest some rival of compressed air should pick up this and run away with it, we might insert a reminder here that all this loss is not necessarily final. It is proper to compare the actual compression line *a o l* with the ideal isothermal line *a c l*, because the volume *c l* is the volume available when the air is put to use. Though at the completion of the compression stroke there is always some of the heat of compression remaining in the air, and its volume is greater than *c l*, that heat is always lost in the transmission or the storage of the air, and the available volume is never practically above *c l*.

After the air has been compressed, and before it is put to use, it is usually transmitted through pipes for some distance, and the friction of the air in its passage through the pipe naturally causes

some loss of pressure. Here again the air-compressor people have—unwittingly, we will say—done more harm than good as regards the interests of compressed air. Formidable tables are in all the air compressor catalogues, showing the loss of pressure due to the friction of air in pipes, and timid investors, seeing them, have another fright. The tables are not dangerous, and are not published primarily for the purpose of frightening people away. They are only intended to suggest the size of pipe most suitable for any given case of transmission. If they tell us truly of the loss of pressure they still fail to tell us that the loss of pressure is not necessarily, or to the same extent, a loss of power. The actual truth is that there is very little loss of power through the transmission of compressed air through proper pipes to a reasonable distance, and the reasonable distance is not a short one. With pipes of proper size and in good condition, air may be transmitted, say, ten miles, with a loss of pressure of less than 1 lb. per mile. If the air were at 80 lbs. gauge, or 95 lbs. absolute, upon entering the pipe, and 70 lbs. gauge, or 85 lbs. absolute, at the other end, there would be a loss of a little more than 10% in absolute pressure, but at the same time there would be an increase of volume of 11% to compensate for the loss of pressure, and the loss of available power would be less than 3%. This illustration is only offered as a simple and convenient one, and not as a sample of the best practice. With higher pressures still more favorable results could be shown.

Having compressed the air and conveyed it to the point where we wish to use it, we may turn again to Fig. 202 and see what we will be able to do with it. The air may be used in various ways with widely different economic results. Having the volume $c\,l$, and using it in a cylinder of suitable capacity, cutting off so as to expand down to one atmosphere before release, the adiabatic line or the lowest expansion line that the air could make would be the line $c\,d$, and the total loss in the use of the air, as compared with the power cost of compressing it, would be the difference between the areas $a\,o\,l\,h$, and $l\,c\,d\,h$, the latter being 66% of the former. The temperature of the air at c, where the expansion begins, being assumed to be 60°, the cooling of the air which always accompanies its expansion will bring the temperature far down the scale when d is reached, d being, of course, the end of the cylinder wherein the expansion takes place. The theoretical temperature of the air at the end of the

stroke would be about $-150°$. The actual temperature is never found as low as the theoretical temperature, because the air receives heat from the cylinder and the walls of the passages with which it comes in contact, but it is usually still cold enough to cause serious inconvenience in practice, and unless some remedy be devised, this cooling of the air is fatal to its employment, entirely regardless of the economy of the case. The air almost invariably contains moisture, the amount varying with the surrounding meteorological conditions, and as the air becomes attenuated and so intensely cold the water in the air is rapidly deposited and frozen in the passages, and soon chokes them up and stops the operation of the engine. So it is useless to try to use compressed air generally, and with general satisfaction, without some means of heating the air immediately before it is set to do its work. For some purposes the air may be and is constantly used without reheating. Hoisting, when done directly, and not by means of a hoisting engine, is, from its intermittent character, not likely to exhibit the freezing phenomenon. Besides the intermittent action of the hoist, giving time for the parts mostly to recover their temperature between the successive hoists, the use of the air in direct hoists is not accompanied with the expansion of the air in the cylinder before its discharge, which expansion, where it exists, is productive of such intense cold. There is occasionally some trouble about operating rock drills on account of their freezing up, but not very often. They are more or less intermittent in their action; they do not generally use the air with much expansion in the cylinder, and the exhaust is sharp and heavy. Many steam pumps are now run by compressed air without its being reheated, never with proper economy, but without their passages freezing up. They generally use low pressures of air, say 20 or 25 lbs., in a great many cases, and they carry a nearly uniform pressure for the whole stroke, so that if there is any trouble about their freezing up it is in the exhaust passage after the air has done its work, and jets of water or some other means of supplying a little necessary heat are sufficient to keep them going. The reheating of the air, where practised, not only brings us out of our trouble about the freezing up of the parts, but it increases the volume of the air and its consequent available power at a very slight expense for the heating. If the volume of air $c\,l$, being now at 60°, be passed through a suitable heater and its temperature raised to

300°, its volume will then be *i l*, instead of *c l*, or .2434 instead of .1666, an increase of volume of about 50%. In practice, to insure a temperature of 300° in the cylinder at the beginning of the expansion, it will be necessary to heat the air considerably above that temperature, say to 400°, as the air loses its heat very rapidly. If now we use this air by filling the cylinder to *i* and then expanding down to *e*, supposing the temperature at *i* to be 300°, the theoretical final temperature will be about zero. The actual temperature, it is pretty certain, will not be below the freezing point, and all our trouble about the freezing of the passages will have disappeared, and the power realized will have been much increased. The use of the reheater being so obviously a necessity we naturally should find it, in various styles, occupying a prominent place in the catalogues of the air-compressor people. Actually and unnaturally, we do not find such a heater in any catalogue, to my knowledge.

As to the cost of reheating the air, it is scarcely within the scope and purpose of this paper to offer any precise data. Such data are easily accessible to any who wish to investigate the subject. I may say, however, that a greater mechanical effect is realized from heat applied to the reheating of compressed air than from any other known application of it, which consideration should urge us to extend its most economical use. There are many purposes, as, for instance, mining and all subterranean operations, where compressed air already *must* be used, regardless of all considerations of economy. There are many others, as, for instance, general hoisting work, where it already *pays* to use compressed air. The extension of its use to the general purposes for which the steam-engine, the gas-engine, or the electric motor are used is distinctly waiting for the general introduction of cheap, efficient, and, above all, "handy" means of reheating the air.

It is quite practicable, by effective cooling of the air during its compression and by reheating it before its re-expansion, to bring the expansion line, *i e*, to enclose an area not less than that enclosed by the compression line *a o*, and then the entire losses will be those attributable to the clearances and to friction. In practice 85% of the initial power has already been realized after transmitting the air to considerable distances.

It was remarked above that the air after compression and transmission might be employed with widely different economic

results. As an instance of how not to do it, I might cite the case, of too frequent occurrence, where air is delivered to a mine for operating rock-drills and other mining machinery, and then air is taken from the same line for operating a pump. This practice would be all right if the pump were adapted to the work to be done and to the pressure of air carried. The pump, however, is generally a common direct-acting steam-pump, which has been obtained without any reference to the economical use of air. As it has probably already been run by steam, or is designed to be run by steam, it calls for a low operating pressure; this being a necessity on account of the condensation of the steam when transmitted through long pipes. Say that the compressed air pipe carries 75 lbs. pressure while the pump only requires 25 lbs. It would be an advantage, in a case like this, to use a pressure reducer in the pipe at a considerable distance from the pump, so that the expansion to the lower pressure required might take place, and the air have an opportunity to recover its temperature and volume before going into the pump. This, however, is seldom attended to, and the required reduction of pressure is effected entirely by the throttle at the instant of admission. The available power, then, when the air is so employed, will be represented by the area *p m n h* as compared with the area *a b l h*, or, at the best, *a o l h*, representing the power that was expended in compressing the air. Then, if we deduct the losses attributable to the useless filling of the large clearances of the common steam-pump and to the leakages which are an universal accompaniment of general loose practice, and consider also the freezing up of the pump which is almost inevitable in such a case, it is little wonder that compressed air is held in little respect. Under circumstances far from the most unfavorable I have found pumps realizing only 15% of the power expended at the compressor, and I have no doubt that there are many pumps being operated, or whose operation is attempted, where not more than 10% of the original power is realized; and even then, when the use of compressed air for operating such pumps is condemned, it is because the pumps freeze up and will not go, rather than on account of the enormous waste of power. From the fact that the duty of a mining pump is a nearly constant quantity, the pump, if properly proportioned and adapted to its work, should be an efficient missionary for compressed air rather than its most malignant traducer.

The word "loss," which we have been using in connection with this subject, is a somewhat incorrect and misleading one. The use of compressed air is for the accomplishment of a desirable purpose, and it is not to be expected that such a purpose can be effected for nothing. The transmission of power is as much to be paid for as the generation of the power. Where water-power is used the means of transmission may be the principal item of cost. Where the difference between the power expended and the power realized is not excessive, that difference is simply a fair price paid for a good service rendered, and there is no loss about it. Where losses do occur in the use of compressed air, they are like the losses which occur in business, and which cut short many a brilliant career. Power is lost simply because it is not saved, and the means of saving are not hard to find nor far to seek. The losses are not necessary nor unavoidable nor without compensation. It is a failure to understand and appreciate this situation which impedes the progress of compressed air.

There is a view of this question of compressed air economy which makes it appear somewhat absurd. It is quite possible, as we have seen, to realize from our motor 80 or 85% of the power received at the air-compressing cylinder. In comparison with this what kind of a model of economy is the steam-engine which is in such universal use? The consumption of water for supplying steam-engines ranges from, say, 12 lbs. per horse-power per hour up to 50 lbs., and away above that, and the consumption of coal per horse power per hour from, say. 1½ lbs. up to 8 or 10 lbs. One may stand in any city or large town in the country, and within a mile of him may be found an engine and boiler using over 400% of the coal per horse-power per hour which is used in the best engines, yet nobody goes around whining about the enormous loss of power in the use of the steam-engine, or tries to discourage anybody from using it, or even seriously tries to promote a better efficiency. It is probable that the air-compressing cylinders of the principal builders of compressors do not vary more than 10% in the economy of their compression. I would hesitate long—and then, of course, refuse—to say that the economy of the steam end of the same compressors showed no more difference than that. To be as explicit as is prudent, I would say that my list of "principal builders" does not exceed four in number.

DISCUSSION.

Prof. D. S. Jacobus.—The author of the paper just read calls attention to the fact that in the use of compressed air for motive power purposes the reheater adds greatly to the work developed.

A short time ago the question arose as to the effect of passing the air through a mass of hot water to perform such heating, as is done in the Mekarski and Hardie compressed air street-car motors. When this is done a portion of the water is evaporated, and the steam thus mingled with the compressed air does work in the motor cylinder. It was found that for the average conditions, shown by tests of the Hardie motor on the Second Avenue Railroad in New York City, where air was admitted to the motor at 120 lbs. pressure above the atmosphere, that the water evaporated into steam did over one-third of the work. In other words, if dry air were brought to the temperature of the hot water the work done would be less than two-thirds that for the moist air. In this case a trifle over one-quarter of a pound of water was evaporated per pound of air.

If the maximum rate of evaporation of water shown in the tests of the Hardie motor was maintained, or about one-half of a pound per pound of air, the work done by the moist air would be twice that for dry air admitted to the motors at the same temperature. The effect of the presence of moisture is to increase the volume of the working fluid entering the cylinder, and to maintain a higher temperature during expansion than that which would occur with dry air. Both of these effects tend to increase the work developed per pound of air. Calculations have been made to show the temperatures of the exhaust for different temperatures of the hot-water bath, and for various amounts of water evaporated. The relative amounts of water and air storage on street-car motors for the best commercial results have also been estimated, together with the distance that motors will travel under given sets of conditions.

The entire subject will probably form a paper to be presented at a later date.

Mr. C. O. Heggem.—Compressed air as a power transmission has been in use at the works of Russell & Co., Massillon, Ohio, for the last six or seven years, beginning with a 600 lb. hoist and increasing from day to day until at present there are in operation 26 five-ton cranes, one cupola stock elevator, and a large number

of small hoists, varying from 400 to 1,000 lbs. capacity. Shears and punches in the boiler shop, where, out of the way of the line shafting, are being operated by compressed air. The accompany-

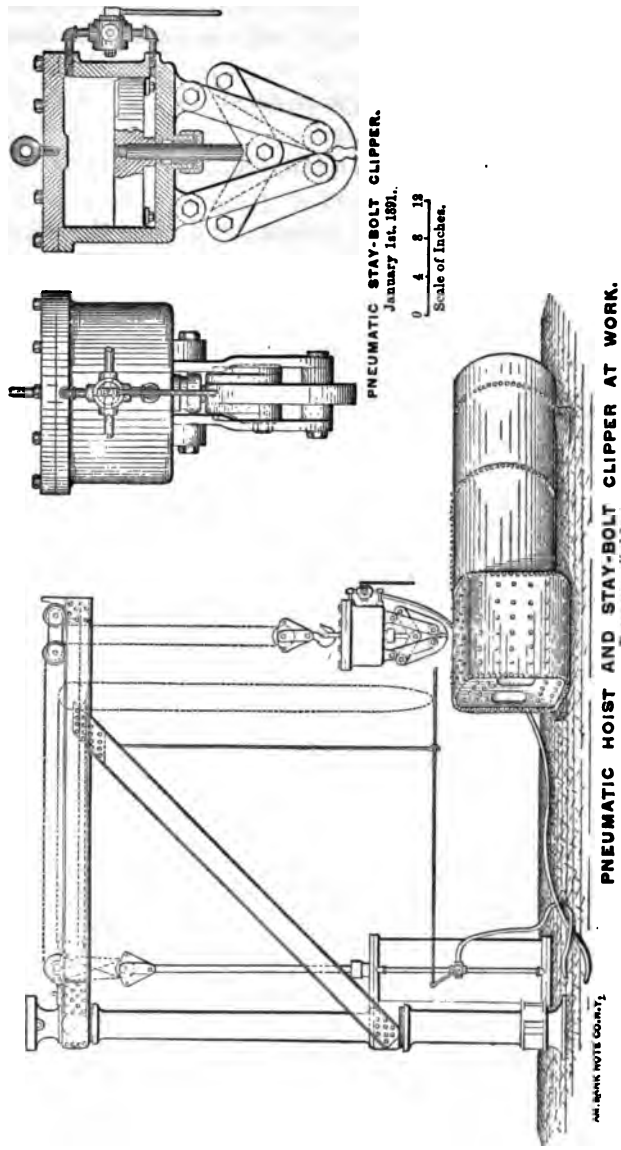


FIG. 208.

ing print (Fig. 203) shows the style of crane and one form of shear as applied, cutting off the stub ends of stay bolts in locomotive

boilers. Here, by the aid of compressed air, two boys may cut off 700 to 800 stay bolts a day, which is far in excess of what could be accomplished with hammer and chisel as formerly used.

It is in the foundry that compressed air is more especially of service, as its elastic action makes it very convenient in drawing patterns, closing flasks, etc.

The air is supplied by three compressors:

One 10 × 13 belt-driven Clayton.

One 7½ × 7 duplex steam-driven Clayton.

And a smaller steam-driven Knowles.

The belted compressor is run constantly during the working hours, and supplies the most of the air required.

The steam-driven Clayton is set to start whenever the air pressure falls below 60 lbs., and, consequently, acts simply as an assistant to the other compressor.

The Knowles is kept as a reserve in case of breakdown.

The air is stored in three reservoirs of 250 cubic feet capacity each, and these reservoirs are placed in such localities of the works where the greatest amount of air pressure is required. Owing to the intermittent use of the air, these comparatively small compressors are able to furnish a sufficient amount and at a cost of something like $3 per day of ten hours.

The piping aggregates about 2,500 ft., and is 2½ inch in the main line, with 1½ and 1¼ inch branches.

Two of the reservoirs are about 1,000 ft. apart, but the pressure in each, as shown by the gauge, is the same.

In laying pipe for compressed air in the ground, care should be taken to go pretty deep in order to insure immunity from frost, as pipes have been known to freeze in this climate two feet below the surface.

Prof. D. S. Jacobus.—In Mr. Richards' interesting discussion of the possibilities of compressed air as a means of transmitting power to motors, there is one statement to the effect that there is no company in the United States which makes a business of applying compressed air, to which I wish to take exception. Those of us who are engaged in sewerage work are acquainted with just such a concern, the Hydro-Pneumatic Sewerage and Water Supply Company; and visitors to the Columbian Exposition last year had an opportunity of seeing twenty-six of its sewage ejectors or pumps working by compressed air, which handled daily a quantity of sewage equivalent to that of a city

of from 100,000 to 400,000 population, according to the attendance. As few mechanical engineers follow sewerage work closely, it may not be out of place to state that these ejectors are automatic devices used in flat places for pumping sewage or water. In such localities it is often difficult to construct gravity sewers of any considerable length, owing to the fact that the trenches soon reach such an excessive depth that the work is difficult and it is hard to discharge the sewage conveniently. In such cases ejectors are employed. The sewage is collected in a number of convenient places from which it is forced in a series of pulsations into main sewers at a higher level. The compressed air for operating the ejectors is supplied through small iron pipes from a single compressor station, often located in the pumping station of the local water-works.

Such appliances are necessarily small; at the Columbian Exposition the capacity of the ejectors was from 60 to 600 gallons a minute. High efficiences are not to be expected, therefore, and those obtained appear very creditable when it is recalled that the liquid pumped is generally unscreened sewage. At Lowestoft, England, some tests made by Prof. W. C. Unwin on a small plant showed an efficiency of nearly 49 per cent. from the steam end of the compressors to the discharge pipe of the ejectors. At Rangoon, under very unfavorable circumstances, a plant having twenty-five ejector stations and seven miles of air pipes showed an efficiency of 33 per cent., although the quantity of sewage handled was so light as to make the test of little value except as an indication of what such apparatus is good for under exceptionally poor conditions. Nevertheless, these figures compare pretty well with those of the pumps of 15 per cent. efficiency mentioned by Mr. Richards, and show that a careful study of all the conditions has enabled compressed air to be used with fair economy, as well as entire sanitary success, in pumping a liquid which is particularly difficult to handle.

Mr. Frank Cawley.—In view of the close study which the author of this paper has given to the subject, as well as the opportunities which he has had at command for observation, I had hoped that he would go more fully into the economic use of compressed air. That air may be used with economy for certain purposes, as compared with other power transmission mediums, I am positive, yet I have never had an opportunity of experimenting in this line.

One of the hardest battles to be fought in selling air-compressing machinery is to overcome the prevalent notion that air is unavoidably a very expensive medium to use, on account of *losses*. Singularly enough, these losses are almost always credited to friction in pipes, which Mr. Richards has rightly shown to be so small a source of loss as to be almost unnoticeable.

That, as usually operated, compressed-air plants are expensive, we must admit; but it is for this very reason that we invite intelligent discussion to the subject, that the methods may be improved.

The author has shown that with average practice the heat and other losses during compression absorb about 20% of the power supplied to the piston of the compressing cylinder. This may be termed the cost of transformation, and as compared with the transformation of hydraulic, caloric, wind, or any other form of energy, into a new form of energy, as rotation of a shaft, for instance, is a favorable result. Few turbines are delivering to their shafts over 70% of the calculated power of the water which passes them; in the double transformation from coal to steam, and from steam to rotation of the engine shaft, the heat units lost are far in excess of those transformed into useful work. Finally, few electricians will guarantee more than 70–75% under average running of the power delivered to the dynamo pulley transformed into electric energy, ready to be applied to any work to which it is suited.

This 20%, good as it may be, is not final; for by the use of more expensive machinery—expense caused principally by machinery being of greater capacity—the air may be compressed more slowly and be more intimately brought into contact with cooling agents, which will prevent a rise of temperature and consequent expansion. Cases also may come up—I have already had a very important one—in which the compressor is operated by water-power, and within 100 feet of the engines to be operated. Here we should let the heat form and jacket our receivers, pipes, etc. to retain it until the engine is reached.

The saving to be gained at this end, however, is small as compared with that at the motors themselves. These for compressed air are usually engines and pumps which have been worn out and "scrapped" under steam, and even though new are entirely unsuited to the economical use of compressed air or steam either. No one would think of comparing an ordinary boiler feed pump,

as regards economy in steam consumption, with a compound Corliss condensing engine, nor should we expect air to make that same pump yield a greater efficiency than it did with steam against the above high-class engine.

For rock drills and percussive coal mining machines we cannot expect much change, for, as in locomotive practice, there are so many considerations calling for large power in small space, and with light weight, that economy is out of the question.

For all pumps and all engines for haulage, hoisting, or other mining or general purposes, air can be and is used with economy. There is no reason—even explosive gas in a mine not always to be excepted—why the air should not be doubled in volume by the addition of heat. There is no reason why this increased volume should not be expanded in a compound engine, or pump, to the atmosphere, and thus could we, with a very small expense for fuel for reheating, realize a handsome return for the power supplied the air piston of our compressor.

To illustrate how badly air is treated I will describe one plant to which I was sent to "make it go," after the trial run had failed.

The compressor I had changed from a 16-inch straight line to a 20-inch with the original 16-inch steam end, etc. It had been set about 80 feet from the boiler, to which it was connected with a 3-inch steam pipe, uncovered throughout its whole length. To make matters worse, at about 25 feet from the boiler a 2-inch pipe was taken out of our main and led 75 or 100 feet away to a hoisting engine which moved from 800 to 1,000 tons of coal a day up the slopes in the mine. Steam on boilers, 60 lbs. and often 45 lbs.

A 3-inch main led into the mine to the first pump, 1,800 feet from the surface; thence 2½-inch, to a second pump, 2,500 feet from the surface; and finally, 2-inch, to a third pump and to the mining machines, 3,600 feet from the surface.

The first pump was an 18-inch × 9-inch × 24-inch "Knowles" previously operated by steam, with the exhaust turned into the suction, as were the other two pumps.

The duty of the first pump was to force the water from old workings and from two lower pumps, to the Cornish pumps which raised it to the surface. When the pump was standing, the pressure in the discharge pipe was 25 lbs.; but when in operation at the usual speed required to get water out in two eight-hour shifts, the pressure was 55 lbs. The discharge pipe was only 5

inches for a 9-inch plunger travelling 160 to 180 feet per minute. These data were given me by the superintendent, who was a very thorough man, and I did not deem it necessary to verify them. The exhaust, of course, was turned into the open air of the pump room, and the pump started. Its condition was then found to be as follows: $\frac{1}{4}$-inch liners had to be placed under the adjusting blocks at the ends of the rocker, in order to secure full and clear exhaust; a $\frac{1}{2}$-inch pipe connected the ends of the 18-inch cylinder, allowing air to blow through it from one end to the other, causing both back pressure and direct loss of air. The gasket under the valve chest had been blown out no one knows how long, between the live steam compartment and the $1\frac{1}{4}$-inch square exhaust ports to the ends of the valve, making an opening at each end over this bridge of 2 inches by $\frac{3}{32}$ inch for air to waste. The pressure on the mains was supposed to be from 60 to 70 lbs. on account of the mining machines below; but this pump, with 55 lbs. on a 9-inch plunger, required but 14 lbs. plus friction requirement on air piston, hence we had to throttle to this extent. The other pumps were respectively a $10 \times 5\frac{1}{2} \times 12$ Cameron, with bad valves, and 50 lbs. water when running, with 10 lbs. standing, and a $12 \times 5\frac{1}{2} \times 12$ fly-wheel—slide valve pump of home construction, with 25 lbs. water running and 10 lbs. standing.

The points which I wish to emphasize here are the bad conditions under which everything had been expected to run. The compressor with a feed pipe an inch too small, naked and robbed of half its steam for other purposes; the air pipe to the mine 3 inches instead of 4 inches; pump No. 1 with its valve gear out of order, a gasket blown out, and a half-inch pipe circulating air from one end of the cylinder to the other; pump No. 2 with extremely bad valves and seats; and all pumps working against heads from twice to five times what their duty called for had the proper discharge pipes been used, and the extent to which air had to be throttled for all.

The agent who sold this compressor was on new ground in an isolated district, and by paying more attention to wild promises than to the idea of successfully and satisfactorily fulfilling them, managed to dispose of three plants of machinery in this district, all of which gave trouble because no regard had been paid to designing or arranging the machinery for the work to be done.

The plant just described might have been made an ideal one, for the mine was perfectly free from gas, and the pipes were in a

continuous run in one of three slopes, the pipe slope, the main slope, and the air or man way. The management of the mine was very progressive, and at that time would have listened to a plan for a representative plant; but, as is too often the case, the agent was a salesman and not an engineer, and his company were satisfied to have him sell their machinery, letting the purchaser place it and run it to suit himself. Unfortunately for them, however, the above agent always made a whole lot of very difficult and sometimes impossible guarantees, which they were expected to make good.

That all plants where compressed air is used are as badly arranged as this, is perhaps too strong, but it is safe to say that the greater number of them are far from what they should be; and if these discussions can start investigation and experiment among those who have the facilities, and lead to a paper on the "Economic use of Compressed Air," the object of the writer in presenting this discussion will be fulfilled.

*Mr. Frank Richards.**—I am glad to have opened to Professor Jacobus the opportunity of mentioning the application of compressed air to sewage disposal. I am aware that there are companies who undertake certain specific applications of compressed air, such as the Sewage and Water Supply Company mentioned, the Pohle Air Lift Pump Company, the Union Switch and Signal Company, the American Pneumatic Tool Company, and many others, each offering an apparatus applying compressed air to a single purpose. Still my assertion holds good that there is no company which undertakes to look after and promote the general application of compressed air and to furnish the variety of appliances required.

It is to be noted that thus far in this country the various established uses of compressed air are of such a character that the question of power economy scarcely applies. Air as yet is little used in motors which take the place of small steam engines, and where the economy of transmission can be computed independently of other considerations. Air is used in mining, tunnelling, caisson work, etc., simply because it must be used for such purposes whatever its cost. Air is now largely used in railroad shops for a wide variety of purposes, simply because there are air-brake pumps lying around or easy to be obtained to furnish

* Author's Closure, under the Rules.

the air, and if economy were very much considered we may be sure that some of those pumps would be replaced by good air compressors.

In the interesting statement by Mr. Heggem of the multifarious use of compressed air in the works of Russell & Co. it is manifestly impossible to compute the power economy of the individual lifts and other machines operated, but the simple fact of their operation costing but $3 per day shows the saving effected in the cost of running the shop, whether the power economy were good or bad. It remains always to be said, that, whatever purpose the air is used for, the economy of power in the compression of it may be properly considered, because any saving there is of necessity a cheapening of the cost of operation.

DLXXXV.*

POWER LOSSES IN THE TRANSMISSION MACHINERY OF CENTRAL STATIONS.

BY WM. S. ALDRICH, MORGANTOWN, W. VA.
(Member of the Society.)

THIS paper had its unexpected origin in correspondence with the Secretary of the Society after Volume XIV. of the *Transactions* had been completed. It was deemed advisable to add some memoranda on the subject of power losses in the transmission machinery of central stations, which had come within the author's experience, and some of which were incompletely referred to in the discussion of Paper No. DXLVI., on the "Performance of Street Railway Power Plants," presented at Chicago meeting (July, 1893), by Messrs. Wm. A. Pike and T. W. Hugo.

The contents of the present paper are outlined below:

(1) A revision of the author's discussion of Paper No. DXLVI.

(2) An analysis of Paper No. DXLVI., with reference to the power losses in the transmission machinery.

(3) A memorandum of the power losses in a turbine plant for an electric railway.

(4) A memorandum of the power losses in a cable plant.

(5) Results of some tests to determine the power losses, in transmitting through a countershaft.

(1) REVISED DISCUSSION OF PAPER NO. DXLVI.

Table XVI., page 1146, Vol. XIV., of the *Transactions* of this Society, of the combined horse-powers of the St. Paul Test, No. 2, from cards taken every ten minutes, cannot possibly show the real variations of power in an electric railway plant. The frictional losses in the transmission machinery are best

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

determined by special tests, before or after the regular run, having this one object in view. In this way the losses in the engine, transmission machinery, shafting and empty cables were determined in a joint test of a cable railway plant, made by Mr. Hermann S. Hering and the author.

It is that which leads me to consider the statement made on pages 1099–1100, regarding the loss in friction as obtained by taking the difference between the mechanical and electrical horse-power. It is true that such is the usual method pursued. In a joint test of an electric railroad, made in 1891, by Mr. Hermann S. Hering and the author, they found that it was impossible to arrive at any satisfactory conclusion as to what became of the motive power generated to drive the dynamos, when throwing the load on and off again as quickly as it is done in an electric railroad. At times there would be as much as 40 H. P. unaccounted for; that is, for a few seconds the dynamo output would fall more than 80 H. P. below the then delivered turbine horse-power; while, at other times, the horse-power delivered by the turbines and the dynamos would be equal. Under light load conditions, which remained steady for about two minutes at one portion of the run, the actual difference between the turbine horse-power and dynamo horse-power, delivered, was 43.8. All at once, while the turbines were decreasing their output, there would be 50 or 60 H. P. thrown off or on the dynamos, and the turbine gates would keep on closing. At other times, the same thing would happen while the turbines were increasing their output, and the turbine gates would keep on opening, causing the difference of power between that delivered by the turbines and that delivered by the dynamos to the line to run enormously high; the maximum difference so observed was 140 H. P., or almost 100 H. P. in excess of that lost in the transmission machinery and dynamo.

Between these extremes one can readily see that the transmission machinery acts in the way of a fly-wheel, storing up the excess of energy on the one hand, and giving out part of its stored energy during a deficiency on the other hand. These great differences of power exist only for a few seconds. The dynamo load does not go down or up to a definite point and stay there, so permitting the turbine to settle to its new load; but, while the turbines are seeking to adjust themselves to the new requirements, the dynamo load has either gone farther in the same

direction, or immediately changed about to the opposite condition. We took the ground that these gross irregularities were largely inherent in a turbine plant.

In writing our report on this test for the *Electrical World* (May 28 to July 30, 1892), we stated that steam-engines would not be at all likely to show the same results of great excesses and deficiencies of output, while the dynamo load continued to vary in its characteristic fashion for street railway purposes. We are greatly surprised, therefore, to find that the tests of Messrs. Pike and Hugo, on the Corliss engines of the Minneapolis and St. Paul electric railway plants, do show up much the same as in our turbine tests referred to. They even note that, when lightly loaded, the difference between the engine and dynamo output is *minus*, showing that the dynamos are delivering more power than the engine, similar to the case we have cited for the turbine plant. While they believe this peculiarity to be due to imperfect (better, *tardy* regulation) we think it is due almost entirely to the fly-wheel action of the transmission machinery. We took the ground that a steam-engine ought to be able to regulate itself more quickly and closely than a turbine, on account of the elastic nature of the working fluid. Probably it would, if it did not have such a drag upon it as heavy transmission machinery and countershafting, in addition to being driven by an incompressible working fluid.

Steam-engines are supposed to be capable of closer and more accurate regulation than turbines, by reason of this very different nature of the working fluid employed. The regulation test of the high-speed automatic engines, noted on page 1116, shows that from a very heavy load to a mere friction load the engine regulated in four revolutions, or one second of time. It is believed that still quicker regulation will come to be the ordinary practice in high-speed automatic engines.

Now, I think it is a very important point to bear in mind that a high-speed automatic engine may be made to meet the rapidly changing loads in an electrical power plant with an almost equal economy. If we take the two papers presented at this meeting—the Laketon test of a triple-expansion pumping engine, in which the load is quite uniform; and the present test of electric railway engines, in which the loads are very irregular—we find much food for thought.

The main point which I wished to bring out in this discussion

is that the high-speed automatic engine, directly connected (or better, directly coupled), is best adapted to meet the sudden and very great variations in electric power plants, while the well-known type of Corliss engine seems best adapted to steady loads.

(2) ANALYSIS OF PAPER NO. DXLVI, WITH REFERENCE TO THE POWER LOSSES IN THE TRANSMISSION MACHINERY.

Table I., following, contains a brief presentation of the principal figures in this connection, from the paper under discussion. It is quite important to notice the increase of frictional losses in the St. Paul Test, No. 2, in which the power is transmitted to the dynamos, through the friction clutches on the counter-shaft. It is probable that there is great frictional loss in the use of certain forms of friction clutches, or in old and ineffective parts. In this connection, also, it is to be especially

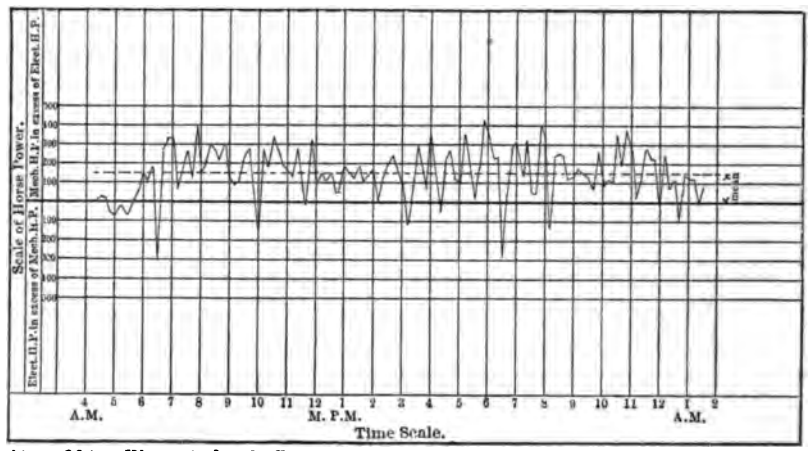


FIG. 204.—Plot of the Differences between Mechanical and Electrical Horse-Powers, for the Minneapolis Test, No. 1.

noted that this No. 2 test was one in which there were belt tighteners used. No doubt a great loss may be accounted for here, also. Both of these causes may have operated to raise the per cent. friction from 12.10 of No. 1, to 17.40 of the No. 2, as tabulated.

Fig. 204 contains a plot of the differences between the mechanical and electrical horse powers, of Fig. 387, opposite page 1098, as given by Messrs. Pike and Hugo for the Minneapolis

Test, No. 1. The characteristic variations are to be especially noted; sometimes the mechanical is in excess, sometimes the electrical. The dotted horizontal line shows the mean value of this difference—the value given on page 1097, for the mean loss by friction, of 149.9 H. P., and as given in Table I, for this test.

TABLE I.

ANALYSIS OF POWER LOSSES IN MINNEAPOLIS AND ST. PAUL TESTS, OF PAPER NO. DXLVI.

No. of Test.	Location of Plant.	Kind of Engine.	How connected to Dynamo.	Ind. H. P. delivered by engine.	Elec. H. P. delivered by dynamos.	Eng. Friction H. P. by Friction Run.	Per cent. Eng. Friction.	Total Friction H. P. = I. H. P. — Elec. H. P	Per cent. Total Friction H. P.
1	Power House, No. 1, Minneapolis.	Tri-cylinder Cond. Corliss.	By Countershaft	1,237.90	1,088.0			149.90	12.10
2	Hill St. Power House, St. Paul.	Tri-cylinder Cond. Corliss.	By Countershaft carrying Friction Clutches.	756.20	624.7			131.50	17.40
2d	Same.	Same.	Same.	859.73				164.43*	19.12
3	21st St. Power House, Minneapolis.	Westinghouse Compounds.	Belted direct to Dynamo.	522.50	434.2	14.27	3.7†	88.30	16.90

(3) MEMORANDUM OF THE POWER LOSSES IN A TURBINE PLANT FOR AN ELECTRIC RAILWAY.

The following particulars of the performance of the turbine power plant of the Neversink Mt. Electric Railway, Reading, Pa., have been taken from the authors' reprint of the published report of the test, as it appeared in the *Electrical World*, May 28 to July 30, 1892. At the special request of this journal, Mr. Hermann S. Hering and the author made a series of tests of this plant, from August 17 to 22, 1891.

* Friction horse-power of engine, transmission machinery, and mechanical and electrical losses in dynamos, obtained by indicating engine with no load on the dynamos.

† Per cent. engine friction, as compared with the maximum horse-power of any one engine.

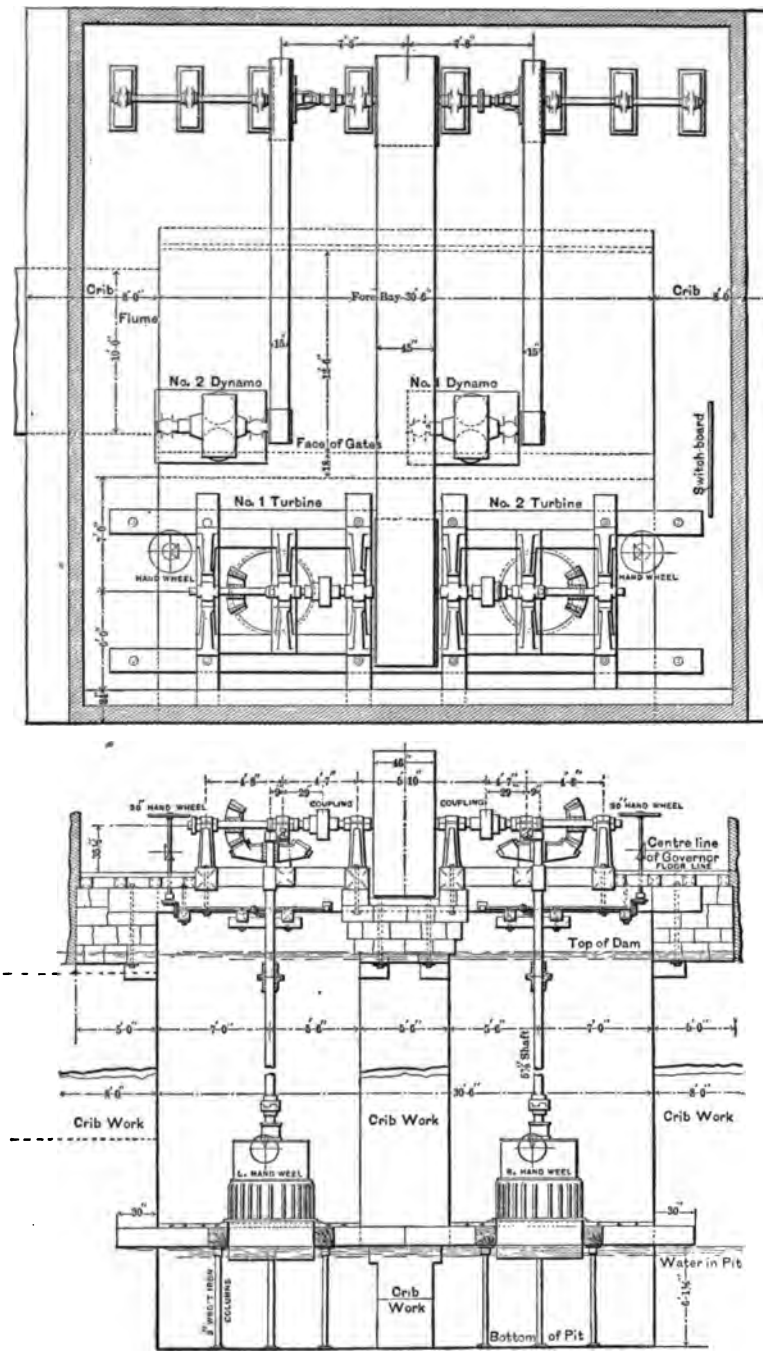


FIG. 205.—Plan and Elevation of the Turbine Plant of the Neversink Mountain Electric Railway, Reading, Pa.

The plan and elevation of the turbine plant are shown in Fig. 205.

Each vertical shaft of the two Hercules turbines transmits its power through a mortise crown and jack gear to the horizontal shaft carrying the main driving pulley. The power is thence transmitted to a counter shaft, from which each dynamo is belted by means of a friction clutch pulley.

Table II. contains the details of the turbine installation, and Table III. of the transmission machinery.

The turbine data for the test was obtained from the Holyoke Machine Company, through whose kindness the authors were furnished with a complete test-sheet of the actual performance of a 45-inch Hercules turbine, of exactly similar design to those here used.

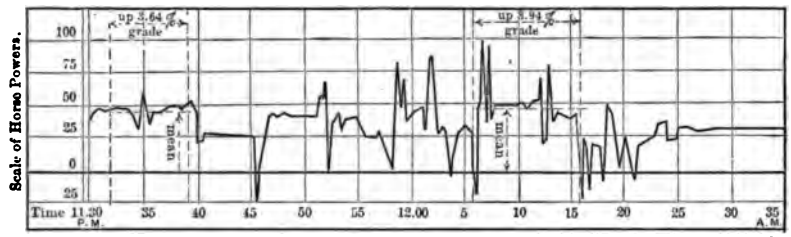
Performance of the Turbine Plant under a Light Load.

In this, known in our report as the "Special Car Test," observations were made to determine the distribution of power, and the several losses from the hydraulic head of the water to the

FIG. 206.—Graphical Log of the Performance of the Neversink Mountain Turbine Plant, under a Light Load.

available power at the car axle, during the run of a single car over the road. The graphical log of the test, so far as the present paper requires, is shown in Fig. 206.

Fig. 207 shows the variations of the losses from the turbine shaft to the dynamo terminals, obtained by taking the instantaneous differences between the turbine horse-power and dynamo output, and plotting on the same time base as Fig. 206. The horizontal dotted line shows the mean (from the integrated means) for the section marked off by the vertical dotted line.



CURVE SHOWING VARIATION OF LOSSES FROM TURBINE SHAFT TO DYNAMO TERMINALS.

FIG. 207.—Plot of the Differences between Mechanical and Electrical Horse-Powers for Reading Test; Light Load.

Table IV. contains the more interesting and important observed and mean values, taken from Fig. 206, and Table V. the summary of the power distribution and efficiencies.

A noteworthy feature of this test is, that the No. 2 turbine was run throughout with a fixed gate at 17.4% proportional part of the full opening of the speed gate. The regulating was entirely done by the No. 1 turbine. Anomalous conditions frequently arise when one of the turbines is thus operated at a fixed gate, as will be readily seen upon an examination of the graphical log, Fig. 206.

Performance of the Turbine Plant under a Heavy Load.

This is designated the "Test of the Power Plant" in our report, and observations were made similar to those referred to in the special car test. The graphical log, as far as relates to the present paper, is shown in Fig. 208, and the variations of the power losses from the turbine shaft to the dynamo terminals in Fig. 209, in which the horizontal dotted line shows the mean (from the integrated means), as before.

Table VI. contains the observed and mean values from Fig. 208, and Table VII. the summary of the power distribution and efficiencies. The time base of the graphical log was divided into two sections, each of which possessed typical groups of changes, and the several values are given for these sections, as well as for the entire run.

In each of these tests there were marked peculiarities in the variations of the power losses between the turbine and the

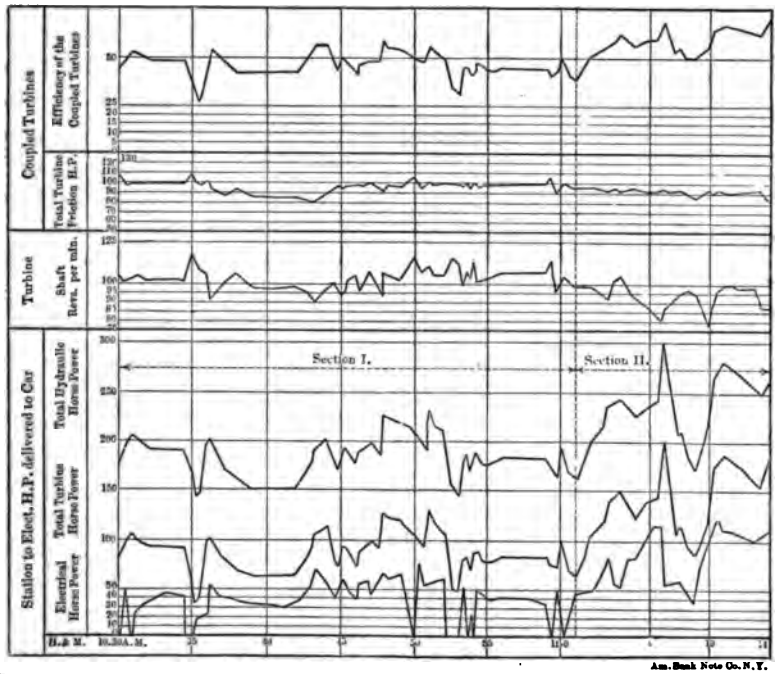
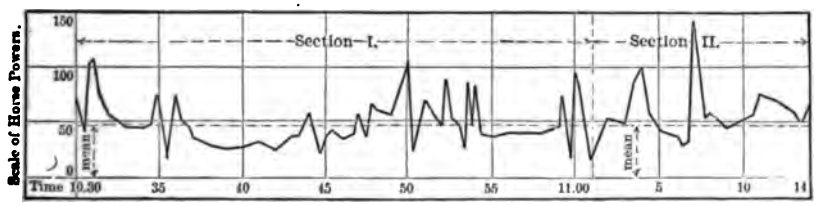


FIG. 208.—Graphical Log of the Performance of the Neversink Mountain Turbine Plant, under a Heavy Load.

dynamo output, as noted in the discussion. It was believed that these were inherent in turbine plants only, but from a



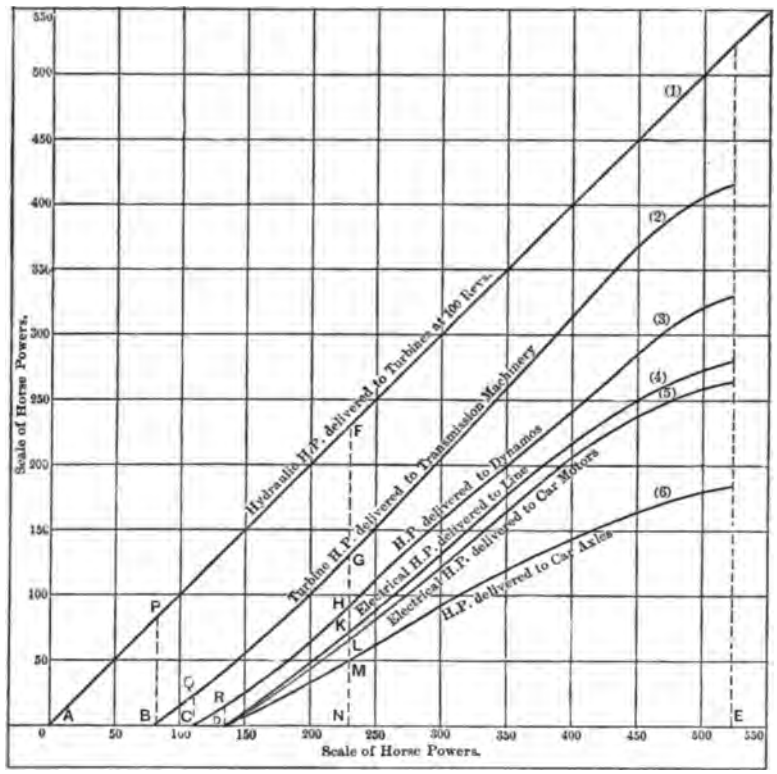
CURVE SHOWING VARIATION OF LOSSES FROM TURBINE SHAFT TO DYNAMO TERMINALS.

FIG. 209.—Plot of the Differences between Mechanical and Electrical Horse-Powers for Reading Test; Heavy Load.

similar plot of the Minneapolis Test, No. 1, given in section (2), it is apparent that the same difficulties are being experienced in electric railway plants driven by Corliss engines.

Average Power Distribution of the Turbine Plant.

The mean values for the several losses in the distribution of the power (mean speed of turbines at 100 revolutions) have been plotted in the curves in Fig. 210. These do not represent the means, integrated or otherwise obtained from few or many observations, nor do they show just the state of affairs at any one moment. They are rather to be considered as general average



CURVES OF POWER DISTRIBUTION FROM HYDRAULIC H.P. TO CAR AXLE H.P.
FIG. 210.

values showing, by simple inspection, the proportional distribution of losses, and the final power available at the car axle for the various horse-powers up to the maximum hydraulic capacity of the station. It would, of course, be possible to obtain such a distribution of the power as is here shown, if the turbine speed could be maintained constant for such a length of time that the station performance would settle down to steady conditions, on

the one hand; while the distribution and load of the cars out on the road should be almost uniformly maintained, on the other hand. The disturbing element in the station performance as here represented is the variable turbine speed; and with regard to the road conditions, it is the varying position of the cars *en route*, and the changing loads on the motors due to varying grades and the somewhat irregular passenger traffic. However, taking all these matters into consideration, the curves convey a very good idea of the relations between the successive losses.

The diagonal line marked (1) represents the H. H. P. delivered to the turbines at 100 revolutions, and for which either the horizontal or vertical scale of horse-powers may be used. The curve (2) is the T. H. P. delivered from the turbine shaft to the transmission machinery. It has been obtained for the several hydraulic horse-powers from the curves which the authors plotted from the data furnished by the Holyoke Machine Company. The horizontal distance, *A B*, Fig. 210, represents the H. H. P. required to drive the two turbines with no output; and that for any other turbine output, as *G N*, is measured by the horizontal distance, *A N*. It is to be carefully noted that the horizontal and vertical distances for any points, as *A B* and *B P*, or *A N* and *N F*, are equal only when referring to points on the diagonal line (1) of H. H. P.

The curve of horse-powers delivered to the dynamos (3) was obtained from a curve of power losses in the transmission machinery at the constant speed given above, and which losses have been set off from the points along (2) for any given horse-power delivered from the turbine shaft. The losses in the transmission machinery were obtained partly from observations made during the test and from calculations based on the dynamo output and efficiency. The point *C*, and its ordinate *Q C*, refer to conditions of the station test, when all the available T. H. P. was absorbed in the transmission machinery, the dynamos being unclutched.

The curve (4) of E. H. P. delivered to the line shows the dynamo output as measured at the switchboard, and its vertical distance, as *H K*, at any point, represents the total mechanical and electrical losses in the dynamos, for that proportional part of the dynamo output. As before, the point *D* and its ordinate *D R* correspond to conditions in which all the available T. H. P. is absorbed in the transmission machinery and dynamo losses,

fields excited but zero amperes at the switchboard. The efficiency of the dynamos at any particular output is shown by the ratio of the respective ordinates of (4) and (3), as $N\ K$ divided by $N\ H$. The losses represented by the vertical distance at any point between (3) and (4), as $H\ K$, comprise the following: Mechanical friction of the journals in the bearing and of the air resistance, 1.34 H.P., and was assumed fairly constant under uniform speed at the different loads; hysteresis, Foucault currents and losses on the shunt fields, 4.22 H.P., also fairly constant under the same conditions; and the variable $C^2\ R$ losses in the armature and the series field coils calculated from resistances with temperature corrections.

The E. H. P. delivered to the car motors, curve (5), has been obtained by setting off vertical distances, as $K\ L$, from (4) which are uniformly 6% of the ordinates of (4) as $K\ N$; this per cent. of line losses being the mean already referred to.

The horse-power delivered to the car axle, curve (6), was drawn by deducting 30% from the measured ordinates of (5), the combined efficiency of the motor and gearing having been taken at 70%. Finally, the ordinates of curve (6), as $M\ N$, represent the horse-power available at the car axle for traction at any particular point within the limits of the preceding curves.

TABLE II.

DATA OF TURBINE INSTALLATION.

Water supply	
Flume, diameter in feet	10
Nominal head of water, in feet	18
Hercules turbines.	
Diameter, in inches	45
Effective area of guide opening, square inches	623
Rated speed, revolutions per minute	110
Commercial rated performance, full gate.	
Rated discharge of each turbine, cubic feet per minute	8,833
Rated efficiency, full gate and rated speed, in per cent	80
Rated horse-power, at this efficiency, each turbine	240
Actual performance at full gate.	
Actual discharge, each turbine, cubic feet per minute	8,588
Actual efficiency, 110 revolutions, in per cent	80.7
Maximum dynamometric horse-power, each turbine	210

Actual performance, about half gate.

Mean position of speed gate, being the proportional part of the full opening of the speed gate in per cent	53.3
Actual discharge at critical speed for this gate, cubic feet per minute	5,528
Efficiency, critical speed 100, in per cent	75.37
Efficiency, rated speed of 110	74.4
Actual dynamometric horse-power of each turbine at the critical speed of 100 revolutions per minute	132.5
Actual dynamometric horse-power of each turbine at the rated speed of 110 revolutions per minute	128.5

TABLE III.

DATA OF TRANSMISSION MACHINERY OF TURBINE PLANT.

Geared and belted connections

Number of teeth on mortised crown gear, vertical shaft	49
Number of teeth on jack gear, horizontal shaft	31
Diameter of main driving pulley, on jackshaft, in inches	108
Diameter of main driven pulley, on countershaft	66
Diameter of dynamo-driving pulleys (friction clutch)	60
Diameter of pulleys on dynamos	26
Main driving belt, extra double, width in inches	45
Dynamo belts, double, in inches	15
Length of main driving belt, in feet	83

Rated speeds.

Turbine shaft (vertical), revolutions per minute	110
Jackshaft over turbines	175
Countershaft	286
Dynamos	650
Speed of main driving belt, feet per minute	4,950
Speed of dynamo belts	4,300

Distance between shaft centre lines.

Between the vertical shafts of the turbines, in feet	16.5
Between the jackshaft and countershaft	30
Between the countershaft and dynamos	20

TABLE IV.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER LIGHT LOAD.

OBSERVED AND MEAN VALUES TAKEN FROM FIG. 3.

Values on the two principal grades, and maximum, minimum, and mean values for entire trip of the Special Test-car, No. 4. August 17 and 18, 1891.

Grade on which reading was taken.	3.64% up.		3.94% up.		3.64% down.	Entire car route.		
Time and time-interval on grade.	11-37-30	11-31-45 to 11-39-30	12-8-0.	12-6-0 to 12-16-0.	12-30.	11-30 to 12-35.		
Conditions.	Steady.	Mean.	Steady.	Mean.	No Load.	Max.	Min.	Mean.
Coupled Turbines.								
Efficiency	49.1	47.4	48.5	46.7	22.3	49.5	−4.88†	34.4*
Total turbine friction H. P.	96.8	98.3	96.3	98.2	97.1	111.5	93.8	98.87
Turbine shaft revolutions per minute	97.0	98.5	97.0	101.5	103.0	122.0	96.0	103.5
Variations of power from hydraulic H. P. at station to electrical H. P. delivered to car.								
Hydraulic H. P.	190.0	187.0	188.2	184.2	124.9	195.0	116.3	150.50
Turbine H. P.	93.2	88.7	91.4	86.0	27.8	95.6	−5.67†	51.75
E. H. P. at station	48.5	45.9	47.6	44.7	0.0	63.9	0.0	.....
E. H. P. at car	46.0	43.1	44.8	42.3	0.0	59.4	0.0	.....

* Calculated from integrated means of horse power curves for entire run. All other values in this column are integrated means.

† No. 2 turbine kept at fixed gate opening of 17.4%. The negative value of the turbine H. P. arises from this turbine driving the No. 1 turbine, at the sudden change of load, indicated ; so that the No. 1 turbine output, the total turbine output, and the efficiency are all negative.

TABLE V.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER LIGHT LOAD.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCIES.

With Special Test-car, No. 4, on the two principal grades, 3.64% and 3.94%, no other car being on the road at that time, August 17 and 18, 1891.

Grade on which reading was taken	*3.64%		**3.94%	
Time and time interval on grade	11-37-30	11-31-35 to 11-39-30	12-8-0.	12-6-0 to 12-16-0.
Conditions.	Steady.	Mean.	Steady.	Mean.
Speed of turbine shaft	97.0	98.5	97.0	101.5
Gate opening, No. 1 turbine	31.7	30.6	31.0	30.2
" No. 2 turbine (fixed gate)	17.4	17.4	17.4	17.4
Speed of car,† miles per hour	12.4	11.43	11.1	10.62
Power delivered:				
Hydraulic H. P. delivered to turbines	190.0	187.0	188.2	184.2
Turbine H. P. delivered to transmission machinery	93.2	88.7	91.4	86.0
H. P. delivered to dynamo	56.1	53.15	55.1	51.9
E. H. P. delivered to line	48.5	45.9	47.6	44.7
E. H. P. delivered to car motors‡	46.0	43.1	44.8	42.3
H. P. delivered at car axles	34.50	31.92	33.7	31.78
Power lost:				
Turbine friction H. P.	93.8	98.3	96.8	98.2
Transmission Machinery Friction, H. P	**37.1**	**35.55**	**36.3**	**34.1**
Dynamo losses	7.6	7.25	7.5	7.2
Line losses	2.5	2.8	2.8	2.4
Car motor, gearing and axle losses	11.5	11.18	11.1	10.52
Total losses, Hyd. H. P. to car axle H. P.	155.5	155.08	154.5	152.42
H. P. available at car axle*	34.5	31.92	33.7	31.78
Total Hyd. H. P. delivered	190.0	187.0	188.2	184.2
Proportional parts of full load:				
Hyd. load, per cent. of full Hyd. H. P.	36.2	35.7	35.9	35.2
Turbine load, per cent. of full turbine H. P.	22.5	21.4	22.1	20.6
Dynamo load, per cent. of full dynamo output	36.2	34.2	35.5	33.3
Motor load, per cent. of full motor output	73.5	68.8	71.5	67.7
Percentage distribution:				
Turbine friction, per cent. loss	.509	.526	.514	.533
Transmission Machinery Friction, per cent. loss	**.195**	**.187**	**.194**	**.185**
Dynamo, per cent. loss	.040	.038	.040	.039
Line, " "	.013	.015	.014	.013
Car motor, gearing and axle losses	.061	.063	.059	.057
Available power at car axle	.182	.171	.179	.173
Total hydraulic power	1.000	1.000	1.000	1.000
Summary of efficiencies:				
Efficiency of No. 1 turbine	58.7	57.4	58.0	55.8
" No. 2 turbine	33.8	32.9	34.0	31.4
" coupled turbines	49.1	47.4	48.5	46.7
Efficiency of Transmission Machinery	**60.2**	**59.9**	**60.3**	**60.4**
" No. 1 dynamo¶	86.5	86.3	86.4	86.1
" line, mean, for given grade	94.8	94.0	94.1	94.8
Efficiency of car motors, from terminals to car axle	75.0	74.2	75.2	75.1
Efficiency from dynamo terminals to car axle	71.2	69.7	70.8	71.2
Efficiency from Hyd. H. P. to line H. P.	25.5	24.5	25.3	24.2
Efficiency from Hyd. H. P. to car axle H. P.	18.2	17.1	17.9	17.2

* On 3.64% grade car started under normal and almost steady conditions. Record ends without stopping the car.

** On 3.94% grade car started from a full stop, stopped twice *en route*, and almost stopped at end of grade.

† Mean speeds on grade are exclusive of stops.

‡ Car equipped with two No. 12 Edison S. R. 25 H. P. motors, style B.

¶ 100 K. W. Edison railway generator, rated at 650 revolutions per minute.

Weight of car alone, 22,000 lbs.; weight of car and load during test, 22,500 lbs.

Plant driven by two 250 H. P. 45-inch Hercules turbines, coupled: one of which, No. 2, is kept at fixed gate opening of 17.4 per cent.

TABLE VI.

PERFORMANCE OF THE TURBINES AND DYNAMOS UNDER HEAVY LOAD.

OBSERVED AND MEAN VALUES TAKEN FROM FIG. 5.

Values on the two principal portions of the run; and maximum, minimum and mean values for the entire run. Aug. 20, 1891.

Section in which reading is taken	I.	II.	Total.
Time base of section, hours and minutes	10.30—11.01	11.01—11.14	10.30—11.14.
Time interval of section, in minutes	31	13	44
Efficiency, maximum	53.0	**69 0**	69 0
" minimum	26.0	39.0	26.0
Mean (from mean H. P's)	46.6	59.4	**51.1**
Turbine friction H. P.			
Maximum	110.0	99.0	110.0
Minimum	82.0	86.0	82.0
Mean (integrated)	97.6	95.1	96.8
Speed of turbine shaft, revolutions per minute.			
Maximum	**113.0**	104.5	113.0
Minimum	99.5	**78.0**	78.0
Mean (integrated)	101.8	93.5	**99.4**
Hydraulic H. P., maximum	230.0	**300.0**	300.0
" " minimum	140.0	162.0	140.0
Mean (integrated)	182.7	234.0	**197.8**
Turbine H. P., maximum	132.0	**208.0**	208.0
" " minimum	37.0	64.0	37.0
Mean (integrated)	85.1	138.0	**101.0**
Electrical H. P., maximum	76.0	**124.0**	124.0
" " minimum	0.0	40.0	0.0
Mean (integrated)	38.5	82.0	**51.3**

TABLE VII.

PERFORMANCE OF TURBINES AND DYNAMOS UNDER HEAVY LOAD.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCIES.

Section in which reading is taken	I.	II.	Total.
Time base of section, hours and minutes	10.30—11.01	11.01 11.14	10.30—11.14
Time interval of section, in minutes	31	13	44
Speed of turbine shaft, revolutions per minute:			
Maximum	**113.0**	104.5	113.0
Minimum	90.5	**78.0**	78.0
Mean (integrated)	101.8	93.5	**99.4**
Gate opening, proportional part of the full opening of the speed-gate, per cent.			
No. 1 Turbine, maximum	33.0	**41.0**	41.0
minimum	**6.5**	8.5	6.5
Mean (integrated)	23.4	25.5	**24.0**
No. 2 Turbine, maximum	44.0	**66.0**	66.0
minimum	**9.5**	16.5	9.5
Mean (integrated)	25.0	38.7	**29.1**
Power Delivered:			
Hydraulic H. P. delivered to turbines	182.7	234.0	197.8
Turbine H. P. delivered to transmission machinery	85.1	138.9	101.0
H. P. delivered to dynamos	50.5	95.8	63.7
E. H. P. delivered to line	38.5	82.0	51.3
E. H. P. " " car motors	31.2	77.1	43.2
H. P. " " car axles	25.4	54.0	33.7
Power Lost:			
Turbine friction H. P.	97.6	95.1	96.8
Transmission Machinery Friction, H. P.	**34.6**	**43.1**	**37.3**
Dynamo losses	12.0	13.8	12.4
Line losses	2.3	4.9	3.1
Car motor and gearing losses	10.8	23.1	14.5
Total losses, Hyd. H. P. to car axle H. P.	157.3	180.0	164.1
H. P. available at car axle	25.4	54.0	33.7
Total hydraulic H. P. delivered	182.7	234.0	197.8
Proportional Parts of Full Load:			
Hydraulic load, per cent. of full hydraulic horse-power	35.1	45.0	38.0
Turbine load, per cent. of full turbine horse-power	20.4	33.4	24.3
Dynamo load, per cent. of full dynamo output	14.4	30.6	19.2
Motor load, per cent. of full motor output	14.5	24.6	17.2
Percentage Distribution:			
Turbine friction, per cent. loss	0.534	0.406	0.489
Transmission Machinery Friction, per cent. loss	**0.189**	**0.184**	**0.189**
Dynamos, in parallel, per cent. loss	0.066	0.059	0.063
Line losses, mean of total length	0.013	0.021	0.016
Car motor, gearing and axle losses	0.059	0.099	0.073
Available power at car axle	0.139	0.231	0.170
Total hydraulic power	1.000	1.000	1.000
Summary of Efficiencies:			
Efficiency of coupled turbines	43.6	59.4	51.1
Efficiency of Transmission Machinery	**59.3**	**69.0**	**63.1**
" dynamos in parallel	76.8	85.2	80.5
" line, mean of total length	94.0	94.0	94.0
" car motors	70.0	70.0	70.0
" from Hyd. H. P. to line H. P.	21.1	35.0	26.0
" from Hyd. H. P. to car axle H. P.	13.9	23.1	17.1

Efficiencies from Hydraulic H. P.	To line H. P.	To car axle H. P.
Efficiency at half load of dynamos, which is 50% turbine capacity, or 57% full Hyd. H. P.	45.0	29.3
Efficiency at full load of dynamos, which is 95% turbine capacity, or 93% full Hyd. H. P.	55.2	36.5

Cars equipped with two No. 12 Edison S. R., 25-H. P. motors, style B. Current supplied by two 100-K. W. Edison railway generators, rated at 650 revolutions per minute.

Plant driven by two 250-H. P., 45-inch Hercules turbines, coupled. Four 22,000-lb. cars out on road, in service during test.

(4) POWER LOSSES IN A CABLE PLANT.

This test was made on the Druid Hill Avenue Cable Plant, Baltimore, Md., jointly by Mr. Hermann S. Hering and the author, March 29, 1892, while associated in the department of electrical engineering at the Johns Hopkins University.

A new plan is employed to drive the cable drums by not having them geared together; but, using in connection with each driving drum, a Whitton compensator. This consists of a simple box-train epicyclic gear. The idler drum is double, one-half of which is mounted loose on the shaft, while the other half is keyed to the same.

The cable drums are placed, one on each side of the rope-drive drum, on the main shaft, and driven through positive clutch couplings. These as well as similar clutches on the engine shaft, are operated by hand, through compound gear.

In the eleven miles of single track, there are fifty-eight single track curves, making 2,900 feet of curve construction in all.

The power house, at the other end of the line, is a duplicate of the Druid Hill Avenue one.

The above particulars are taken from the *Street Railway Journal*, March, 1891, which contains a full and illustrated description of the "Baltimore Cable Road" and the Druid Hill Avenue power house.

The test was made under the ordinary working conditions of the road, the usual daily traffic and operations not being interfered with in the least during the normal running time of 24 hours. The plant consisted of two Wetherill-Corliss type of engines, rated at 414 H.P., at 60 revolutions, 80 lbs. boiler pressure, and $\frac{1}{5}$ cut-off. Rope-driving is used to transmit the power from the engine shaft to the cable-driving shaft. Only one engine was run during the test. It was fitted with pantograph reducing motion, and two Thompson indicators. The method of indicating the engine was to take simultaneous cards from each end of the cylinder every 2 minutes. The pencil was kept on for a single revolution only, and the same card was used for five consecutive observations, at 2-minute intervals each. New cards were put on every ten minutes, so that each card contained a record of five single diagrams from one end of the cylinder. Sometimes the pencil, at one end of the cylinder, would be left on the card for 3 or 5 minutes, while the single

cards were taken from the other end as usual. On comparing the two cards it was found that the mean obtained from the five single cards was practically the same as that from the one continuous card. The method therefore followed, in taking the cards and averaging them, gave a satisfactory value of the mean indicated horse-power of the engine during the interval. The sum of the areas of the five single diagrams was divided by 5, giving the mean area, which divided by the mean length gave the mean effective pressure for the 10-minute interval.

The frictional losses of the engine, drum, and cables were determined by running the engine with only that part of machinery attached, of which it was desired to obtain the friction horse-power. It will be noticed that the sum of the individual friction horse-powers of the large and small cables is not equal to that required to drive the two together. This shows an increase in the friction of the transmission machinery and through the rope drive, a result which is to be expected when the transmitted load is increased and the two cable-driving shafts are operated as one.

The power necessary to drive the cable, under no load, and at various speeds, was obtained by a special test; from the plotted results of this run it appears that the cable friction horse-power varies almost directly with the speed at which the cable is running.

It is the extent of curved track and the steep grades on the road, which no doubt make the efficiency of the cables so low; for, as will be seen by referring to the summary of efficiencies, only 45.9% of the power delivered to both cables is available for use in propelling the cars.

The data and particulars involving the power losses in the transmission machinery are tabulated below:

TABLE NO. VIII.

DATA AND PERFORMANCE OF ENGINE AND CABLE.

Dimensions of Engine.

Diameter of cylinder..... 28 inches.
Stroke of piston..... 5 feet.
Piston stroke to release (proportional part of stroke)....100.0 stroke.
Piston stroke to exhaust closure .....head, 92%; crank, 92%
Clearance from card .....head, 2.7%; crank, 3.0%

Dimensions and Data of Transmission and Cable Machinery.

Driving drum, on engine shaft..... 8 feet diameter, 8 feet 4-inch face.
Main shaft drum.....24 " " " " "
Number of ropes in rope-drive.....32
Main shaft, between clutch couplings.....18 inches diameter.
Main shaft, between clutch couplings and
the cable-driving drums.....15 " "
Cable drums, down-town.....10 feet 6 inches diameter
" " up-town ....16 " diameter.
Cable speeds, down-town..... 7½ miles per hour.
" " up-town.....11½ " "

Data from Test.

Date of test ....March 29, 1892.
Duration of test, normal running time of engine ..21 hours, 52 minutes.
Mean boiler pressure, pounds per gauge.....80.96 lbs.
Ratio of boiler to initial engine pressure.....88.3%
Mean speed of engine.....60.53 revolutions per minute.
Mean piston speed.....605.3 feet per minute.
Average mean effective pressure..... { head end, 20.6 lbs. / crank " 22.4 "
Mean total indicated horse-power ....241.7 H.P.
Mean back pressure in ports..... { head end, 1.91 lbs. / crank " 1.43 "
Mean back pressure of compression ..... { head end, 21.0 lbs. / crank " 22.0 "

TABLE IX.

SUMMARY OF POWER DISTRIBUTION AND EFFICIENCY OF CABLE PLANT.

Power Developed and Delivered.

Horse-power developed, cars on, maximum 6.30 P.M.....342.2 H.P.
" " " minimum 5.04 A.M. .....156.0 "
" " " mean (of run) .....241.7 "
Horse-power developed without cars on, but with both cables, drum,
and engine friction, maximum at start.....174.6 "
Horse-power developed without cars on, but with both cables, drum,
and engine friction, minimum.....148.8 "
Horse-power developed without cars on, but with both cables, drum,
and engine friction, mean .....152.35 "
Horse-power developed without cars on, but with down-town cable,
drum, and engine friction, mean.....124.2 "
Same, but with up-town cable, drum, and engine, mean..... 69.0 "
Same, but with drum and engine only, mean ..... 47.45 "
Same, but with engine only (engine uncoupled), mean..... 29.15 "
Horse-power delivered by engine shaft.....212.55 "
Horse-power delivered by main-driving shaft from clutch couplings.194.25 "

Horse-power delivered by both cables to cable car grips	89.35 H.P.
Friction horse-power of both cables and additional shafting required by same, mean	104.9 "
Friction horse-power of both cables, alone, mean	98.3 "

Power Lost.

Engine friction horse-power	29.15 H.P.
Drum friction	18.30 "
Up-town cable friction	21.55 "
Down-town cable friction	76.75 "
Shafting, due to mean load on both cables	6.60 "
Total friction horse-power	152.35 H.P.
Mean available horse-power delivered to cars	89.35 "
Mean total indicated horse-power developed by engine	241.70 H.P.

Summary of Percentage Distribution of Power.

Percentage lost in engine friction	12.10%
" " drum friction	7.60%
" " up-town cable friction	8.90%
" " down-town cable friction	31.80%
" " shafting (due to use of both cables)	2.70%
Percentage available power at cars	36.90%
Total power developed by engine	100.0%

Summary of Efficiencies.

Mechanical efficiency of engine	87.9%.
Efficiency of rope-drive transmission, from engine shaft to main-shaft clutch couplings	**91.4%.**
Efficiency of both cables and cable-driving drums, from main-shaft clutch coupling	46.0%.
Efficiency of the cable plant, from the indicated horse-power of engine to power available at cars	36.9%.

(5) POWER LOSSES IN TRANSMITTING THROUGH A COUNTERSHAFT.

These tests were made to determine the losses in the usual method of driving dynamos; namely, through two belted connections and a countershaft. They were made jointly by Mr. Clement R. Jones and the author, in the Mechanical Engineering Department of the West Virginia University, in March, 1894.

Two sets of tests were made:

(1) To determine the engine friction.

(2) To determine the transmission machinery friction.

(1)—*In the engine friction tests*, 30-minute runs were made, and indicator cards and speed of engine taken every minute. The value of the engine friction under load is the mean of four 30-minute runs, with water-cooled Prony brake and platform scales for measuring the power, with minute observations, as in the first case. The results are given below.

TABLE X.

DATA AND PERFORMANCE OF ENGINE.

Dimensions of Engine.

Engine, Wheelock, rated horse-power, at 75 revolutions, 80 lbs. initial pressure, gauge, ¼ cut-off	40 H. P.
Diameter of cylinder	10¾ inches.
Stroke of piston	30 inches.
Crank shaft, crank bearing	4½ inches diameter, 11 inches long.
" " outboard bearing	4½ inches diameter, 10 inches long.
Band-wheel; diameter on crown	96⅝ inches; face, 17 inches.

Engine Performance.

Engine running free,—column (*a*).
Engine running under Prony brake load,—column (*b*).

	(*a*)	(*b*)
Time interval of run, minutes	30	30
Corrected mean boiler pressure	76.00	75.70
Mean revolutions, per minute	74.93	71.18
Mean indicated horse-power	2.77	17.18
Mean Prony brake horse-power	0.00	14.57
Engine friction, horse-power	2.77	2.61
" " per cent	100.00	15.19
Mechanical efficiency of the engine, per cent.	0.0	84.81

(2)—*In the transmission machinery friction tests*, two separate runs were made, one without and the other with leather covering on dynamometer pulley, as noted in the following table under (*c*) and (*d*) respectively. The power was transmitted through a countershaft and absorbed by an Alden dynamometer, as shown in Fig. 211, on which the principal dimensions are also given.

It was possible to maintain a very steady load on the engine by means of the Alden dynamometer, and its constant was carefully determined and allowed for in the calculations. It is a 32-inch 50-H. P. dynamometer, especially designed and con-

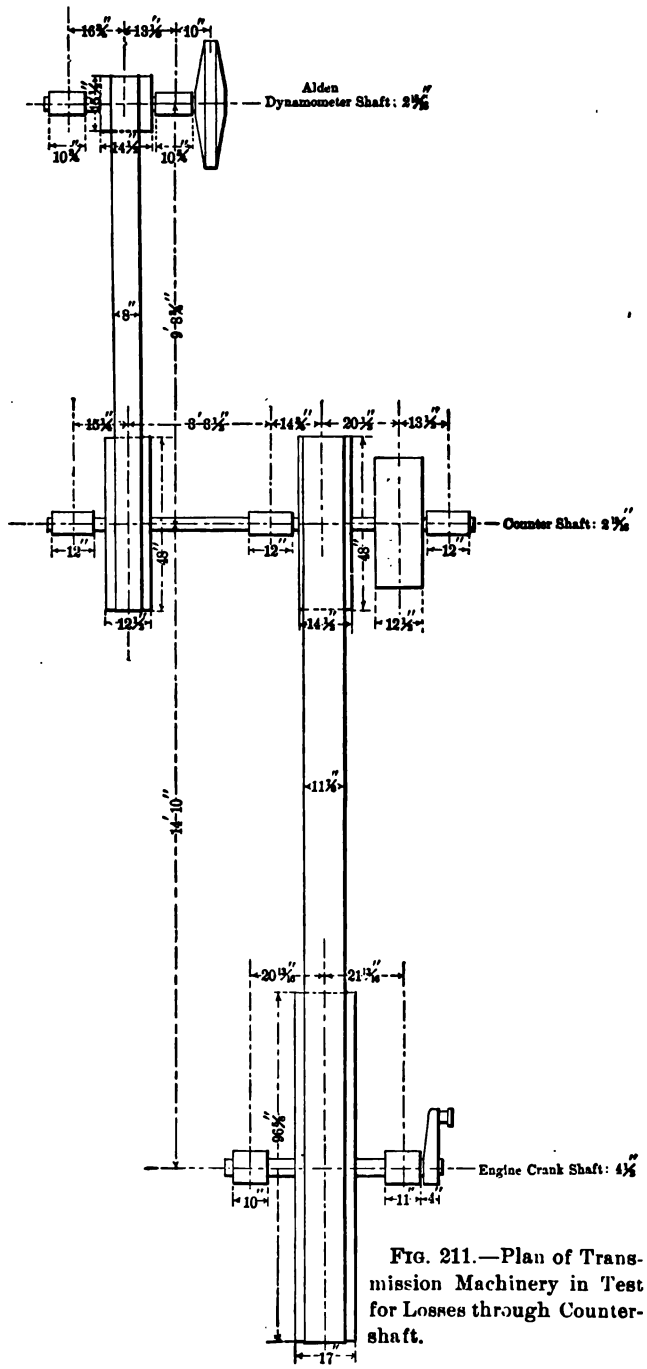


FIG. 211.—Plan of Transmission Machinery in Test for Losses through Countershaft.

structed for use with high-water pressure at high speeds; but, during this test, it was worked far below its limit, as small powers only were absorbed. Its position in the test is that which a dynamo would occupy in the ordinary electric light plant, driven by a slow-speed engine. It is believed that tests of this character will serve to show the ordinary losses in such installations.

Indicator cards and speeds of engine and dynamometer shafts were taken every two minutes. The engine friction horse-powers, tabulated for (*c*) and (*d*), were taken from a curve plotted from the previously recorded results, on the usual assumption that the engine friction varies directly as the speed, regardless of the load.

TABLE XI.

DATA AND PERFORMANCE OF THE TRANSMISSION MACHINERY.

Dimensions and data.

Band-wheel of engine—diameter on crown	96⅝ inches; face, 17 inches.
Countershaft pulleys, driven—diameter on crown	48 " " 14½ inches.
" " driver " "	48 " " 12½ inches.
Dynamometer pulley (*c*) bare cast-iron: diameter on crown	15¼ inches; face, 14½ inches.
Dynamometer pulley (*d*) covered with Shultz patent leather covering: diameter on crown	15½ inches; face 14½ inches.
Main driving belt, light double, oak-tanned	11⅞ × ½ inch.
Dynamometer belt, single, oak-tanned	8 × $\frac{7}{32}$ inch.

Transmission Machinery Performance.

Plain cast-iron pulley on dynamometer,—column (*c*).
Shultz patent leather covering on dynamometer pulley,—column (*d*).

	(*c*)	(*d*)
Time interval of run, minutes	60	30
Corrected mean boiler pressure	59.29	64.5
Mean revolutions per minute, engine	67.0	68.25
" " " " dynamometer	402.60	417.06
Calculated speed of dynamometer, ratio	6.34	6.234
" " " " speed	424.8	425.5
Loss in speed, from engine to dynamometer shaft	22.2	8.44
Slip of belt, per cent. of calculated speed	5.23	1.98
Tension at which belts were laced and run, average of initial, and final tensions on belts—pounds per square inch:		
Main driving belts	40	80
Dynamometer belt	75	135
Mean indicated horse-power	19.122	13.21
Engine friction, horse-power (from plotted curve)	2.50	2.54

Horse-power delivered to transmission machinery	16.622	10.670
Dynamometric horse-power absorbed	15.275	9.960
Horse-power lost in transmission machinery	**1.347**	**0.710**
Frictional loss, per cent., engine	13.08	19.23
Frictional loss, per cent., transmission machinery	**8.10**	**6.65**
Efficiency, per cent., engine	86.92	80.77
Efficiency per cent., transmission machinery	**91.90**	**93.35**

DLXXXVI.*

TESTS OF A SMALL ELECTRIC RAILWAY PLANT.

BY JESSE M. SMITH, DETROIT, MICHIGAN,
(Member of the Society.)

THIS paper is designed to give in a concise form the results of a test made in October, 1893, on the power plant of the Wyandotte & Detroit River R. R., which is a suburban road running from Detroit to Trenton, Michigan, 10.5 miles long.

The plant contains two tandem compound condensing engines rated at 150 H.P. each, and guaranteed for a maximum load of 225 H.P., made by the Phoenix Iron Works Co., Meadville, Pa. Each engine is belted direct to a Westinghouse railway generator of 150 E.H.P. The vacuum is obtained by a Conover air pump and condenser belted directly to the engine shaft. The boiler feed pump is driven from the air pump shaft. Steam is supplied by two Manning vertical boilers. One engine and dynamo only were used during the test. It will be noted at the outset that each engine and dynamo is designed to deliver an average of 150 H.P., but that during the test the average load on the engine was only 70 I.H.P., and the maximum 141 I.H.P. The best conditions of economy were therefore not realized. The test was made under commercial conditions for the owners of the plant. All the machinery had been run about three months, and no adjustments by the makers had been made since it was first put in. No special care had been taken to get very close regulation either of the engine or generator. One boiler was fired with coal, and used exclusively to supply steam for the fuel atomizer of the other boiler. "Straight crude oil" was used exclusively as fuel for the boiler, which supplied steam to the engine, and no steam was used from that boiler for other purposes. Indicator cards were taken at intervals of 5 minutes during the 17⅓ hours of the test.

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* Presented at the Montreal meeting, June, 1894, of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

A Tabor indicator was applied directly to each end of each cylinder, and all operated by the same mechanism. Indicator cards were taken simultaneously from the four indicators by two persons on a signal given at the proper time, without regard to the load on the engine. The speed of the engine was determined by a continuous revolution counter which was directly connected to the shaft of the engine. The variation of speed was shown by a tachometer belted directly to the engine shaft, and constantly in service. It was carefully compared with the revolution counter when on constant speed, and was made to agree with the counter. The speed of the dynamo was taken at four different times by a revolution counter held in the end of the shaft for five consecutive minutes each time. Three readings gave 604 revolutions, and one reading gave 603 revolutions per minute. The speed of the air pump was taken in the same way.

The steam gauges on the boilers were compared to a standard gauge from the University of Michigan, and found to be correct. These gauges are connected to the boiler near the bottom, so that they show the pressure due to a head of water of about 10 feet in addition to the steam pressure. The boiler gauges therefore show about 5 pounds more pressure than the pressure of the steam in the boiler. The standard gauge, when placed on the steam pipe of the engine just above the throttle valve, showed about 5 pounds less pressure than the boiler gauge, so that there was practically no loss of steam pressure between the boiler and engine. The vacuum gauge was compared with a standard gauge from the University of Michigan, and showed a vacuum 1.5 inches higher than the standard. All the data given below have been properly corrected to agree with the standard instruments. The temperatures were taken by standard thermometers from the University of Michigan.

The amperes were measured by a Weston standard ammeter connected in series with the station ammeter. The volts were measured by a Weston standard volt meter connected to the bus bars of the switch board. Volt and ampere readings were taken every ten seconds by a different person at each instrument, the time being given by a third person.

The water used by the boilers was taken from the overflow of the air-pump, and weighed in two barrels on two scales, and was pumped from the barrels by the regular feed pump worked from the shaft of the air pump. The feed water passed through a

closed heater with brass tubes, the heater being connected with the exhaust pipe between the engine and condenser. The fuel oil was stored and fed by the Snell water pressure system. The amount of fuel oil fed to the atomizer was measured by measuring the number of gallons of water which flowed into the storage tank.

The test proceeded continuously without mishap of any importance, and I was greatly assisted by Messrs. Weber, Becker, Bird, Clark, Ames, and Hamper, engineering students from the University of Michigan, and also by H. E. Whitaker, junior member of the Society, who also worked up the indicator cards and made the calculations of results.

The dimensions of the plant and data, and the results are given in the following tables:

DESCRIPTION OF PLANT.

Boilers.

Kind	Manning Vertical Tubular.
Number	2
Size	Known as 61-inch.
Rated capacity	150 H.P.
Inside diameter of fire-box	6 feet 0 inches.
Height above grate	3 feet 6 inches.
Number of tubes	184
Outside diameter of tubes	$2\frac{1}{2}$ inches.
Total length of tubes	14 feet 10 inches.
Length of tubes below water-line	10 feet 7 inches.
Heating surface in fire-box	89.2 square feet.
Heating surface in tubes below water-line	1272.7 square feet.
Heating surface in tubes above water-line	511.1 square feet.

Engine.

Kind	Phœnix Iron Works Co. Tandem Compound Condensing.
Diameter of high-pressure cylinder	$11\frac{1}{16}$ inches.
Diameter of low-pressure cylinder	$20\frac{1}{32}$ inches.
Stroke of piston	15 inches.
Piston valves on both cylinders, both controlled by shaft governor.	
Number of fly-wheels	2
Diameter of fly-wheels	$78\frac{3}{8}$ inches.
Face of fly-wheels	$16\frac{3}{8}$ inches.
Weight of rim of each fly-wheel	2,445 lbs.

Air-Pump.

Kind	Conover belted to engine.
Size	No. 7.
Diameter of piston	15 inches.
Stroke of piston	7 "
Revolutions per minute	67

Boiler Feed-Pump.

Kind	Single-acting plunger driven from shaft of air-pump.
Diameter of plunger	3 inches.
Stroke	6 "

Dynamo.

Kind	Westinghouse 4-pole Railway Generator.
Size	No. 1.
Rated capacity	111.9 kilowatts.
Rated capacity	150 E. H. P.
Standard speed	625 rev. per min.
Speed when tested	604 " "

Cars.

Maker	Jones.
Length	16 feet.
Weight	14,500 lbs.
Number of seats	22
Kind of motors	Westinghouse.
Number of motors	2
Rated power of each motor	30 E.H.P.
Number of cars in use	3

Road.

Length	10.5 miles.
Kind of rail	T
Weight of rail per yard	52 and 56 lbs.

Supported on cross ties.

Ties covered with sand and resting on sand, except from Wyandotte to Trenton, which is stone ballasted.

SPEED OF ENGINE AS CONTROLLED BY GOVERNOR.

Revolution per minute by *revolution counter*, average for 8 minutes with dynamo belt off	238 revolutions.
Same, with dynamo running with brushes on, but main circuit open, average of 6 minutes	238 "
Average revolutions per minute on regular work during 362 minutes	234.3 "

Percentage of variation of revolutions	1.58%
Maximum revolutions per minute, as shown by *tachometer*, dynamo producing 0 amperes, engine developing 14.3 I.H.P.	239 revolutions.
Minimum revolutions per minute, dynamo producing 180 amperes, engine developing 165 I.H.P.	227 "
Maximum revolution, as shown by *tachometer* during *regular run*	237 "
Minimum revolutions, as shown by *tachometer* during *regular run*	233 "
Average revolutions, as shown by *tachometer* during *regular run*	234.5 "
Total variation	4 "
Percentage of total variation	1.7% "

DISTRIBUTION OF POWER USED.

Power consumed by friction of engine, air-pump, and boiler feed-pump, with main belt off	9.22 I.H.P.
Friction of engine, air and feed-pumps, and dynamo, with brushes off	11.34 "
Friction of dynamo and belt	2.12 "
Power consumed by engine, air and feed-pumps and dynamo, with brushes on and main circuit open	14.34 "
Power required to charge fields of dynamo	3.00 "
Rated capacity of engine	150 "
Rated capacity of dynamo	150 E.H.P.
Percentage of engine capacity required by friction of engine, air, and feed-pumps $\frac{9.22}{150} =$	6.15%
Maximum power developed by engine, as shown by indicator cards, during economy test	141.4 I.H.P.
Minimum power developed by engine, as shown by indicator cards, during economy test	21.27 "
Average power developed by engine, as shown by 208 sets of indicator cards, during economy test	70.1 "
Maximum volts developed by dynamo	520 volts.
Minimum " " "	480 "
Average " " "	501 "
Maximum amperes developed by dynamo	200 amperes.
Minimum " " "	4.7 "
Average " " "	67 "
Average watts delivered by dynamo	33,567 watts.
Average electrical horse-power delivered by dynamo	45 E.H.P.
Average indicated horse-power delivered to pulley of dynamo, estimating friction of armature shaft to be the same as friction of belt	59.8 I.H.P.
Average commercial efficiency of dynamo $\frac{45}{59.8} =$	75.25%

Average number of cars in use during test	2.89 cars
Number of passengers on cars during 17½ hours	1,014
Number of single trips of cars	64
Average number of passengers on cars per single trip	15.2
Average number of passengers on cars at any one time (estimated)	8
Weight of ten persons at 140 lbs. each	1,400 lbs.
Weight of cars	14,500 "
Total weight of cars and persons	15,900 "
Average weight in motion	45,950 "
Average electrical horse-power per 1,000 lbs. of weight moved	0.98 E.H.P.
Average horse-power developed by engine per 1,000 lbs. of weight moved	1.52 I.H.P.
Average watts required per car	11,615 watts.
Average electrical horse-power per car	15.54 E.H.P.
Average horse-power developed in engine per car	24.25 I.H.P.
Average watts delivered by dynamo per horse-power developed in engine	478.8 watts.
Average watts delivered by dynamo per horse-power delivered to belt of dynamo	557.3 watts.

RATE OF SPEED OF CARS.

Length of road	10.5 miles.
Time of round trip, including 2 minutes stop at each end and at 16 railroad crossings	90 minutes.
Average speed per hour, including all stops, 21 miles in 1.5 hours	14 miles per hour.

Running time, allowing 2 minutes at each end 4
1¼ minutes at each railroad crossing 4

90 − 8 = 82 minutes.

Average speed between stops = 21 miles in $\frac{82}{6}$ = 1.366 hours = 15.38 miles per hour.

MILEAGE OF CARS.

2 cars, 11½ round trips each	23 miles.
1 car, 10½ " " "	10½ "
Total number round trips	33½ "
Total car miles—33½ x 21	703.5 "
Average miles run by each car	234.5 "

TIME OF SERVICE OF CARS.

2 cars, 11 round trips, 1.5 hours each	33 hours.
1 car, 10 round trips, 1.5 hours each	15 "
Car hours	48

Time required by each car to get to end of road to commence its trip and get back to power house	¾ hour.
Total time of cars away from power house—	
2 cars, 16.5 + 0.75	34.50
1 car, 15 + 0.75	15.75
Total car hours	50.25
Time of test	17.33 hours.
Average number of cars in use during test $\frac{50.25}{17.33}$	2.89

ECONOMY TEST.

Duration of test	17⅓ hours.
Average horse-power developed in engine as shown by 208 sets of indicator cards taken at 5 minute intervals	70.1 I.H.P.
Average horse-power delivered by dynamo as shown by 6,209 sets of readings taken at 10 second intervals	45. E.H.P.
Net average steam-pressure in boiler by half hour readings	111.8 lbs.
Average vacuum by half hour readings	25.2 inches.
Steam used by engine in 17⅓ hours	29,829.6 lbs.
Steam used by engine per hour	1721.1 "
Steam used by engine per horse-power per hour	24.55 "
Steam used by fuel oil jet in 17⅓ hours	5064.5 "
Steam used by fuel oil jet per hour	291.5 "
Steam used by engine and fuel oil jet per hour	2008.8 "
Percentage of steam used by fuel oil jet	14.5%
Weight of oil per gallon at 61° Fahr	6.57 lbs.
Fuel oil used to raise steam-pressure before starting, 5 gal	32.9 "
Fuel oil used for engine during test of 17⅓ hours, 251.7 gal	2308.3 "
Fuel oil used per hour for engine, 20.25 gal	133 "
Fuel oil used per horse-power per hour for engine, 0.29 gal	1.9 "
Additional fuel oil which would be required to maintain oil jet for 17⅓ hours, 59.71 gal	393.3 "
Additional fuel oil which would be required to maintain oil jet for one hour, 3.43 gal	22.58 "
Total fuel oil that would be required during day, 416.4 gal	2734.5 "
Average temperature of water flowing from air-pump	91.8° Fahr.
Average temperature of feed-water entering boiler	136.6° "
Rise of temperature of feed-water due to heater	34.8° "
Percentage of fuel saved by heater	3.06%
Fuel oil required to evaporate 1000 lbs. of water from 126.6° Fahr., into steam at 112 lbs. pressure	11.79 gals.
Water evaporated from 126.6° Fahr. into steam at 112 lbs. pressure per gallon of fuel oil	84.81 lbs.
Ditto per pound of oil	12.9 "
Kind of fuel oil used	Straight Ohio Crude.
Kind of fuel oil atomizer	Snell
Area of steam outlet of atomizer	0.07 sq. in.

Which is same as a round hole 0.3 inches diameter.

It will be noted that 14.5% of the total steam generated was used by the fuel oil atomizer. The atomizer was not properly proportioned for the work, and did not receive enough air. The flame was red and evidently not economical. The atomizer and furnace have since been re-arranged with much better results.

In this engine the piston valves which distribute the steam to both high and low pressure cylinders are controlled by the governor. The effect of this is clearly shown by the following three sets of indicator cards, which are representative of the cards taken during the economy test. With each set is a card from the receiver between the high and low pressure cylinders taken at the same time as the others of the set.

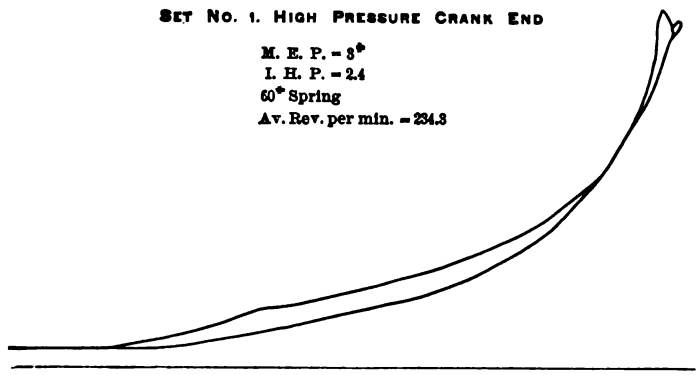


FIG. 212.

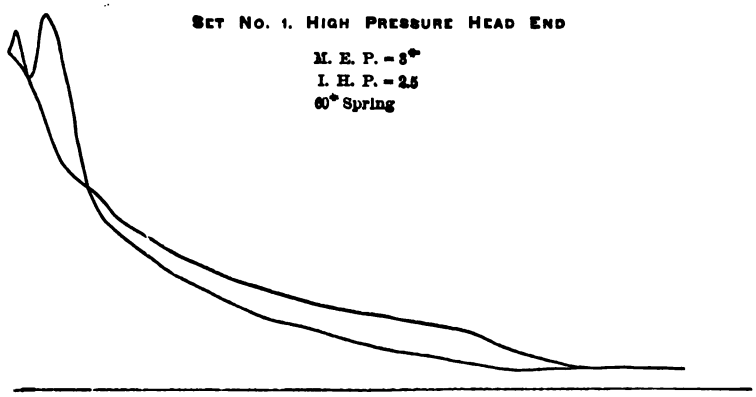


FIG. 213.

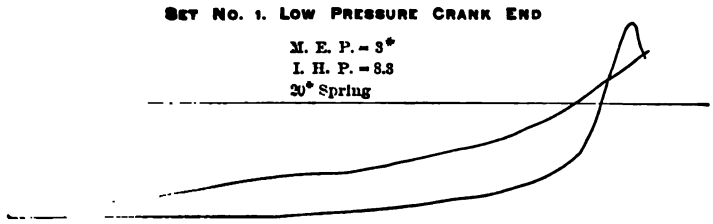


FIG. 214.

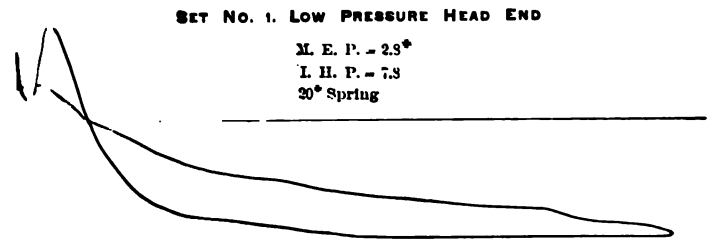
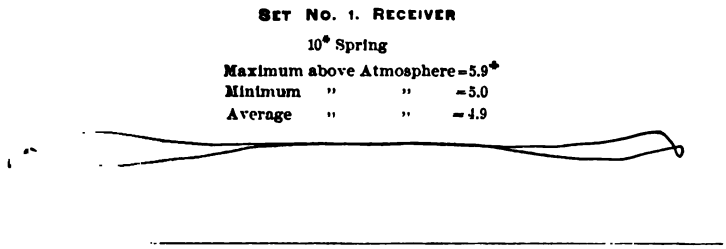


FIG. 215.



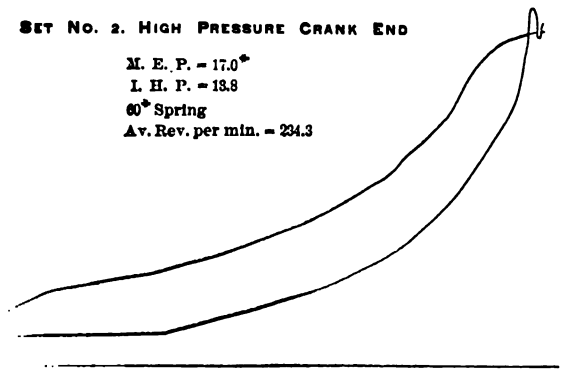


FIG. 216.

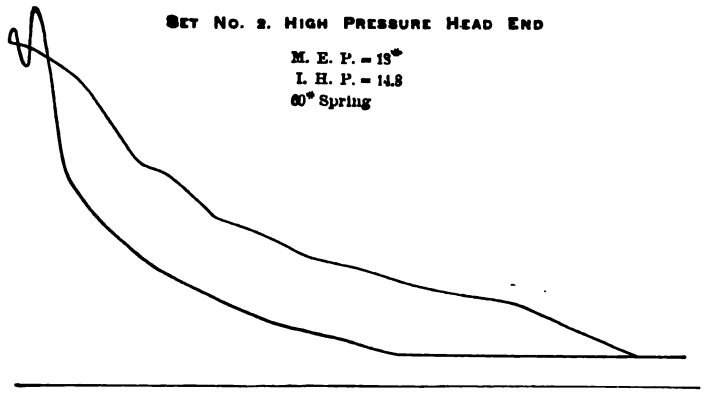


FIG. 217.

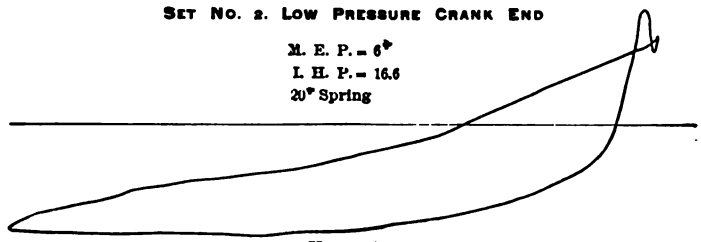


FIG. 218.

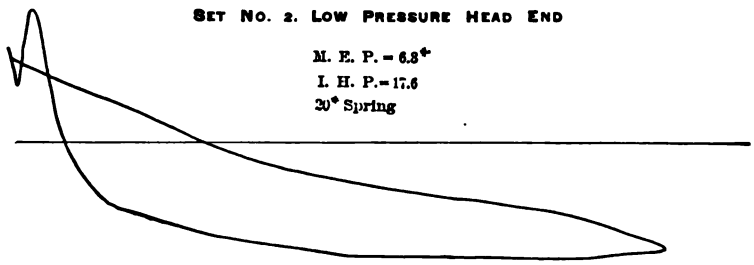


FIG. 219.

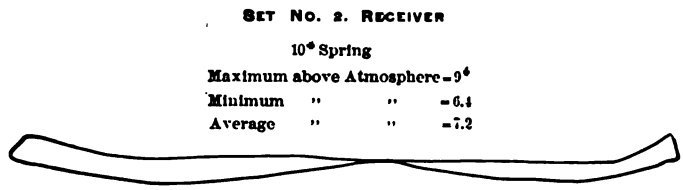


FIG. 220.

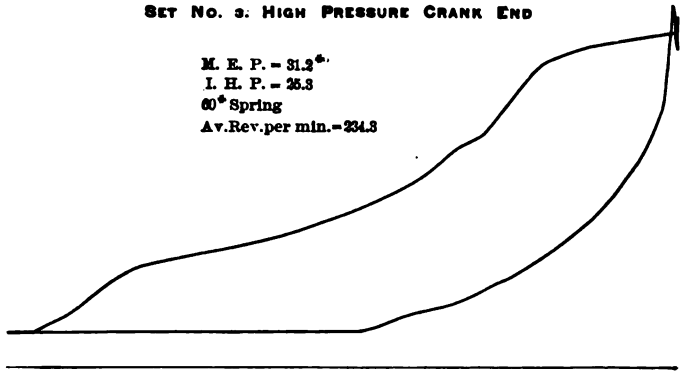


FIG. 221.

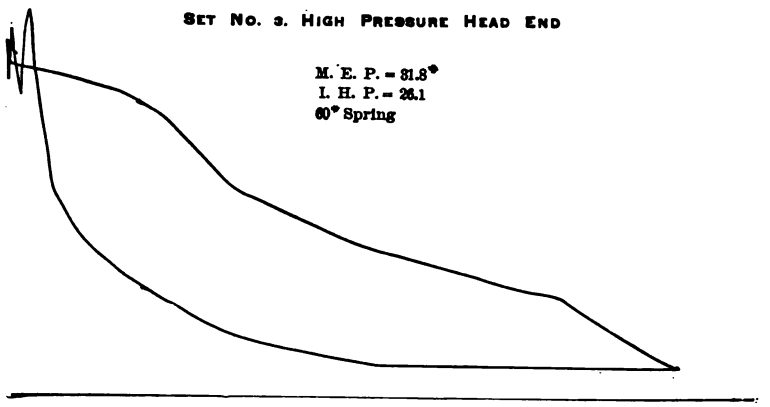


FIG. 222.

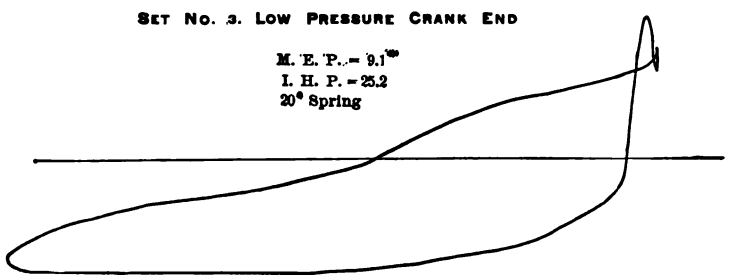


FIG. 223.

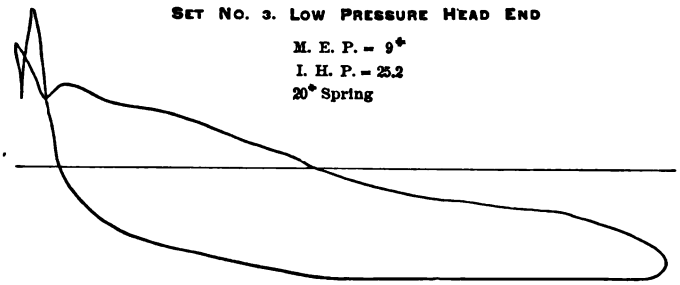


FIG. 224.

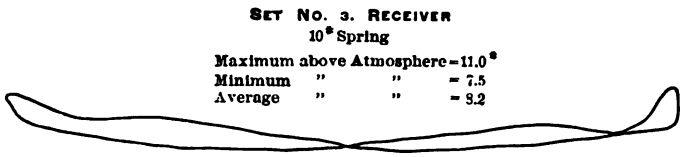


FIG. 225.

Each set shows the horse-power developed in each end of each cylinder, as well as the variation of pressure in the receiver.

The results are given in the following table:

	Set No. 1.	Set No. 2.	Set No. 3.
I. H. P. in H. P. Cylinder.			
Crank end.....	2.4	13.8	25.3
Head end.....	2.5	14.8	26.1
I. H. P. in L. P. Cylinder.			
Crank end.....	8.3	16.6	25.2
Head end..	7.8	17.6	25.2
Total I. H. P.....	21.0	62.8	101.8
Maximum pressure in receiver...	5.9	9.0	11.0
Minimum " " "	4.0	6.4	7.5
Average " " "	4.9	7.2	8.2

It will be noted that while this is a condensing engine the high pressure cylinder does a good share of the work even on the lightest loads, and the receiver pressure does not vary greatly either during a single revolution, or during a change of load of from 21

to 101.8 horse-power. While the load changes 485%, the average receiver pressure only changes 167%, and even when the engine ran with the belt off and developed 9.2 horse-power, there was no negative work in either end of either cylinder. These results, which I believe to be highly important, both for economy and close regulation, are due to the valves of both cylinders being controlled by the governor.

The variations in load in a plant of this kind are not only extreme but rapid.

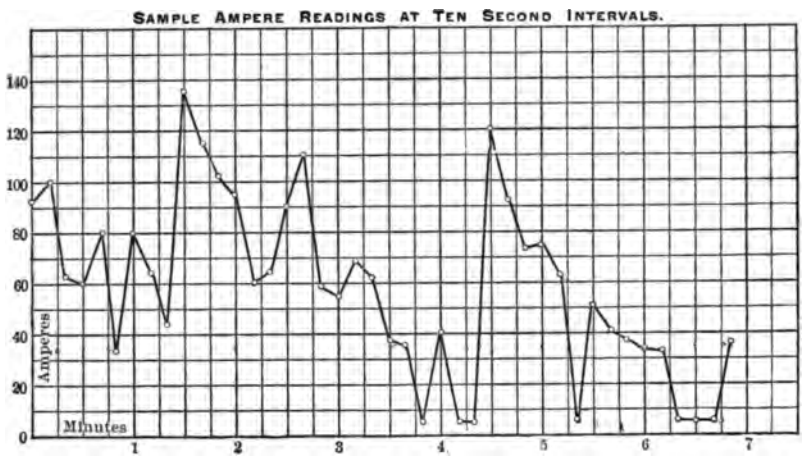


FIG. 226.

Fig. 226 shows a sample of the ampere readings during about seven minutes, at intervals of ten seconds.

The variation is from 5 to 134 amperes, and in one interval of ten seconds the load increased 115 amperes, which equals 120 indicated horse-power.

During the regular operation of the plant and with these extreme variations in load, the needle of the tachometer never left the space enclosed by the 230 and 240 revolution marks.

DISCUSSION.

Mr. C. W. Baker.—There is one feature of this paper which should not pass without some discussion. The author says: "It will be noted that 14.5% of the total steam generated was used by the fuel oil atomizer." Now, if we were firing coal under a boiler and it took one-seventh of all the steam the boiler made to run an

automatic stoker to put the coal in the furnace, we would think that a very expensive method of firing. It ought to be brought out whether such an expenditure of steam for spraying is a general rule in using fuel oil under boilers, or whether this record is something exceptional.

Mr. Jesse M. Smith.—In reply to Mr. Baker I will say that the amount of steam required by the atomizer is certainly excessive, and as a result of this test the furnace, as well as the atomizer, was changed, so that I am satisfied it is now giving much better results. This discussion emphasizes the fact that when oil is used as fuel, account is not often taken of the amount of steam required by the atomizer, and which is a dead loss so far as the power of the boiler to produce useful work is concerned.

While I think 14.5% is more steam than should be necessary for properly atomizing the oil, still a very considerable quantity is always necessary, and in careful estimates this must of necessity be taken into account.

DLXXXVII.*

EFFECT OF VARYING THE WEIGHT OF REGENERATOR IN A HOT-AIR ENGINE

BY G. W. BISSELL, AMES, IOWA.
(Junior Member of the Society.)

WHEN the writer was connected with Cornell University, as instructor in experimental engineering, he noticed that experiments with a Rider Compression Pumping Engine gave data showing a lowering of temperature of the jacket-water in its course through the engine, instead of an elevation of temperature, as he expected. Many experiments, conducted carefully and with special pains in the calibration of thermometers, and in their use by interchange of position, confirmed the phenomenon.

The experiments which are herein described, and of which the data are given, were made to find out what effect, if any, varying the amount of metal in the regenerator would have upon the phenomenon referred to.

The Rider Compression Hot Air Pumping Engine is too well known to require description. The fuel used consisted of wood used for starting the fires and hard coal. The amount used in each run was determined as in the standard method of steam-boiler trials. All of the water pumped passed through the jacket of the displacing cylinder, except a small stream which was carried through the jacket of the working cylinder. Both streams of water were united beyond the pump and delivered to the weir, which measured the water pumped. The head against which the pump worked was made by throttling the discharge, and was measured by a pressure-gauge in the usual way. The indicator was attached to the regenerator chamber, and its drum received motion from the working cylinder. The speed was obtained by hand counting. The thermometers used for determining the temperatures of the jacket-water were eighteen inches

* Presented at the Montreal meeting, June, 1894, of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

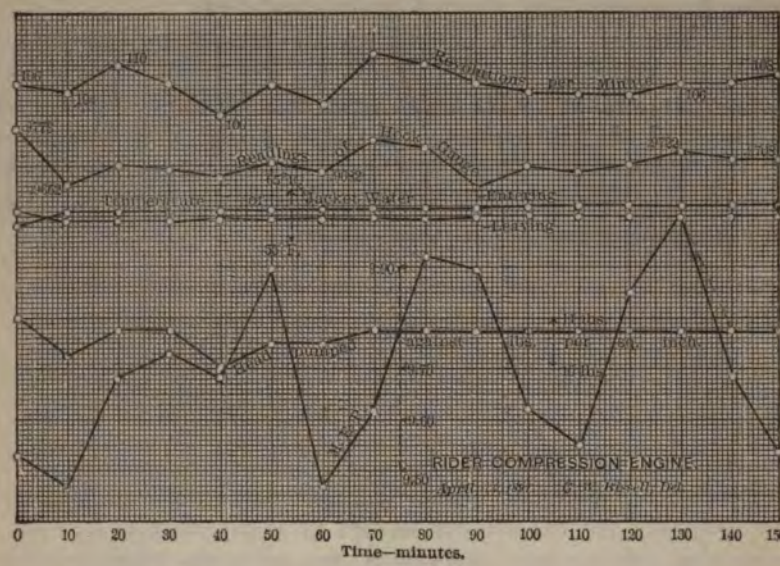
long, graduated to 1/5° Fahr., and carefully compared, besides being frequently interchanged during each test.

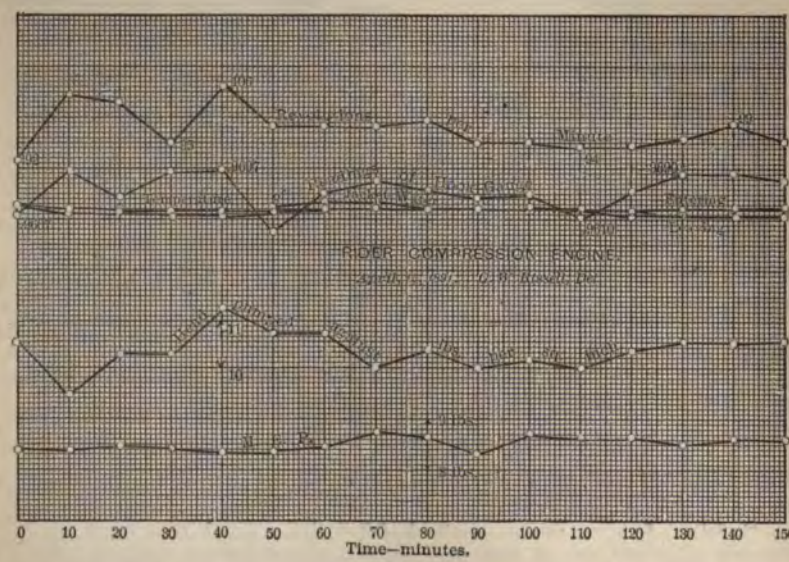
The data and results are appended. Graphical logs and a graphical record of the final results are also given. The observations were taken by Messrs. Clarke and Heilman, then senior students in the laboratory, the latter being now instructor in the laboratory. The writer gave close supervision to the experiments here reported, and frequently verified the readings of the temperatures of the jacket-water.

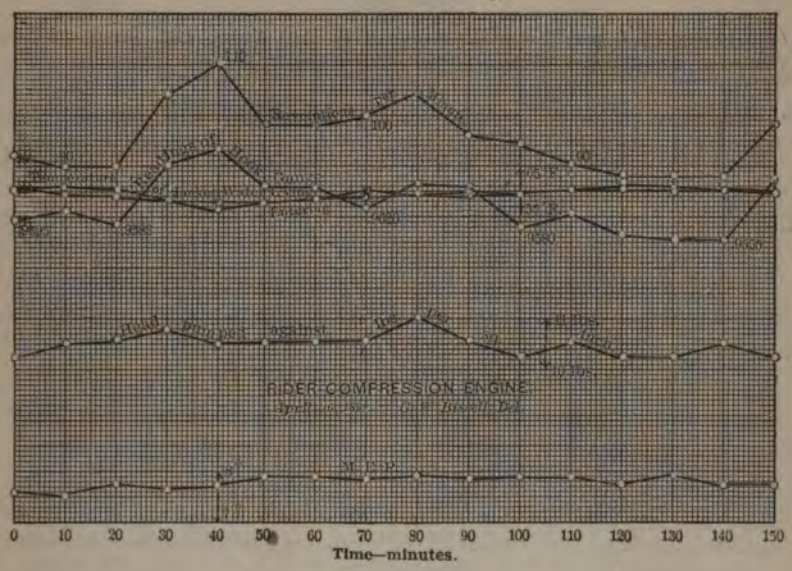
TEST NO. 1.

All of regenerator in ; weight of same, 20.75 lbs. Date, April 14, 1891. Duration, 2 hours 30 minutes. Zero of weir, 0.7853 foot. Weight of wood, 6.2 lbs. Weight of coal, 4.63 lbs.

No.	Time.	Weir Reading.	Head, lbs.	M. E. P.	Temperatures.				Revs. per Minute.
					Room.	Water at Weir.	Jacket.		
							Entering.	Leaving.	
1	3.20	0.9773	11.00	9.53	77	86.8	58.2	61.2	106
2	3.30	0.9662	10.25	9.47	..	87.5	60.9	59.0	104
3	3.40	0.9703	10.75	9.68	..	88.0	60.9	58.9	110
4	3.50	0.9691	10.75	9.73	..	88.4	60.7	58.9	106
5	4.00	0.9675	10.00	9.68	76	89.3	60.9	59.5	100
6	4.10	0.9704	10.50	9.89	..	89.9	61.1	59.2	106
7	4.20	0.9682	10.50	9.47	..	90.1	61.1	59.5	102
8	4.30	0.9748	10.75	9.62	..	89.6	61.3	59.3	112
9	4.40	0.9733	10.75	9.92	7	89.8	61.2	59.2	110
10	4.50	0.9653	10.75	9.89	..	89.9	61.4	59.8	106
11	5.00	0.9695	10.75	9.62	..	90.4	61.5	59.8	104
12	5.10	0.9687	10.75	9.55	77	90.5	61.7	59.9	104
13	5.20	0.9700	9.85	9.85	..	90.8	61.9	59.9	104
14	5.30	0.9722	10.75	10.00	..	91.0	61.9	60.0	106
15	5.40	0.9712	10.75	9.68	77	91.0	61.9	59.9	106
16	5.50	0.9708	10.75	9.53	..	91.0	61.9	59.9	108
Mean ..	.....	0.9703	10.66	9.69	.....	89.62	61.16	59.62	105.9


RIDER COMPRESSION ENGINE
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150
Time—minutes.


RIDER COMPRESSION ENGINE
0 10 20 30 40 50 60 70 80 90 100 110 120 130 140 150
Time—minutes.



TEST NO. 2.

Two-thirds of regenerator in; weight of same, 13.5 lbs. Date, April 15, 1891. Duration, 2 hours 30 minutes. Zero of weir, 0.7853 foot. Weight of wood, 8.75 lbs. Weight of coal, 6.5 lbs.

No.	Time.	Weir Reading.	Head, lbs.	M.E.P.	Temperatures.				Revs. per Minute.
					Room.	Water at Weir.	Jacket.		
							Entering.	Leaving.	
1	3.00	0.9600	10.50	8.39	83	78.4	62.9	62.9	92
2	3.10	0.9690	9.50	8.34	..	79.9	62.2	61.5	104
3	3.20	0.9642	10.25	8.49	..	81.6	61.8	61.2	103
4	3.30	0.9690	10.25	8.39	..	82.5	61.8	60.9	95
5	3.40	0.9697	11.25	8.32	79	83.9	61.7	60.7	106
6	3.50	0.9573	10.75	8.33	..	85.0	61.8	61.4	98
7	4.00	0.9648	10.75	8.39	..	86.1	62.2	63.2	98
8	4.10	0.9673	10.00	8.62	78	87.3	62.3	62.9	98
9	4.20	0.9659	10.35	8.62	..	84.6	62.2	62.2	99
10	4.30	0.9638	10.00	8.32	..	84.5	62.2	62.3	96
11	4.40	0.9645	10.15	8.70	78	85.7	62.5	62.3	95
12	4.50	0.9610	10.00	8.57	..	86.8	62.6	62.4	94
13	5.00	0.9660	10.35	8.62	..	88.0	62.9	62.0	94
14	5.10	0.9690	10.50	8.47	..	88.6	62.9	61.6	96
15	5.20	0.9689	10.50	8.62	78	89.2	62.9	61.7	99
16	5.30	0.9678	10.55	8.62	..	89.4	62.9	61.7	96
Mean ..	.....	0.9655	10.35	8.49	79	85.0	62.3	61.9	97.7

TEST NO. 3.

One-third of regenerator in; weight of same, 6.75 lbs. Date, April 16, 1891. Duration, 2 hours 30 minutes. Zero of weir, 0.7853 foot. Weight of wood, 8.0 lbs. Weight of coal, 6.0 lbs.

No.	Time.	Weir Reading.	Head, lbs.	M. E. P.	Temperatures.				Revs. per Minute.
					Room.	Water at Weir.	Jacket.		
							Entering.	Leaving.	
1	2.20	0.9595	10.35	7.56	78	93.0	65.5	65.8	92
2	2.30	0.9610	10.50	7.48	..	93.6	64.8	66.2	90
3	2.40	0.9583	10.55	7.78	..	94.5	64.5	65.7	90
4	2.50	0.9700	10.75	7.65	..	95.9	63.8	63.5	104
5	3.00	0.9734	10.50	7.72	78	96.4	63.0	61.6	110
6	3.10	0.9660	10.55	7.87	..	92.1	63.0	63.3	98
7	3.20	0.9658	10.55	7.87	..	92.4	63.7	63.8	98
8	3.30	0.9620	10.60	7.80	..	93.5	63.9	64.9	100
9	3.40	0.9667	11.00	7.87	78	94.0	64.5	64.3	104
10	3.50	0.9653	10.55	7.80	..	93.7	64.6	64.9	96
11	4.00	0.9580	10.25	7.87	..	94.4	64.5	64.5	94
12	4.10	0.9603	10.50	7.87	76	95.5	65.3	68.5	90
13	4.20	0.9565	10.25	7.72	..	90.9	65.8	65.4	88
14	4.30	0.9555	10.25	7.87	..	90.3	65.7	65.3	88
15	4.40	0.9555	10.50	7.65	75	90.4	65.2	66.5	88
16	4.50	0.9693	10.25	7.68	..	89.6	64.7	65.0	98
Mean ..	.....	0.9626	10.49	7.75	77	93.1	64.5	64.9	95.5

RESULTS.

Diameter of working piston, 6.75 inches; diameter of pump plunger, 1.5 inches; length of stroke of working piston, 0.792 foot; length of stroke of pump plunger, 0.713 feet.

Test No.	1	2	3
Date	4—14—1891	4—15—1891	4—16—1891
Duration	2½ hours	2½ hours	2½ hours
Weight of regenerator, lbs.	20.75	13.50	6.75
Head pumped against, feet	24.51	23.80	24.12
Weir reading	0.9703	0.9655	0.9626
Weir zero	0.7853	0.7853	0.7853
Head over weir, feet	0/1850	0.1802	0.1773
Water delivered, cubic feet per second	0.0215	0.0202	0.0193
Temperature of water at weir, Fahr.	89.62	85.00	93.1
Effective horse-power	0.0590	0.0542	0.0528
M.E.P.	9.69	8.49	7.75
Revolutions per minute	105.9	97.7	95.45
Indicated horse-power	0.880	0.711	0.632
Mechanical efficiency, per cent.	6.79	7.62	8.20
Coal consumed per hour, lbs.	2.84	4.00	3.68
Coal per I.H.P., per hour	3.22	5.62	5.82
Initial temperature of jacket-water	61.16	62.30	64.5
Final temperature of jacket-water	59.62	61.9	64.9
Rise in temperature of jacket-water	−1.54	−0.4	+0.4

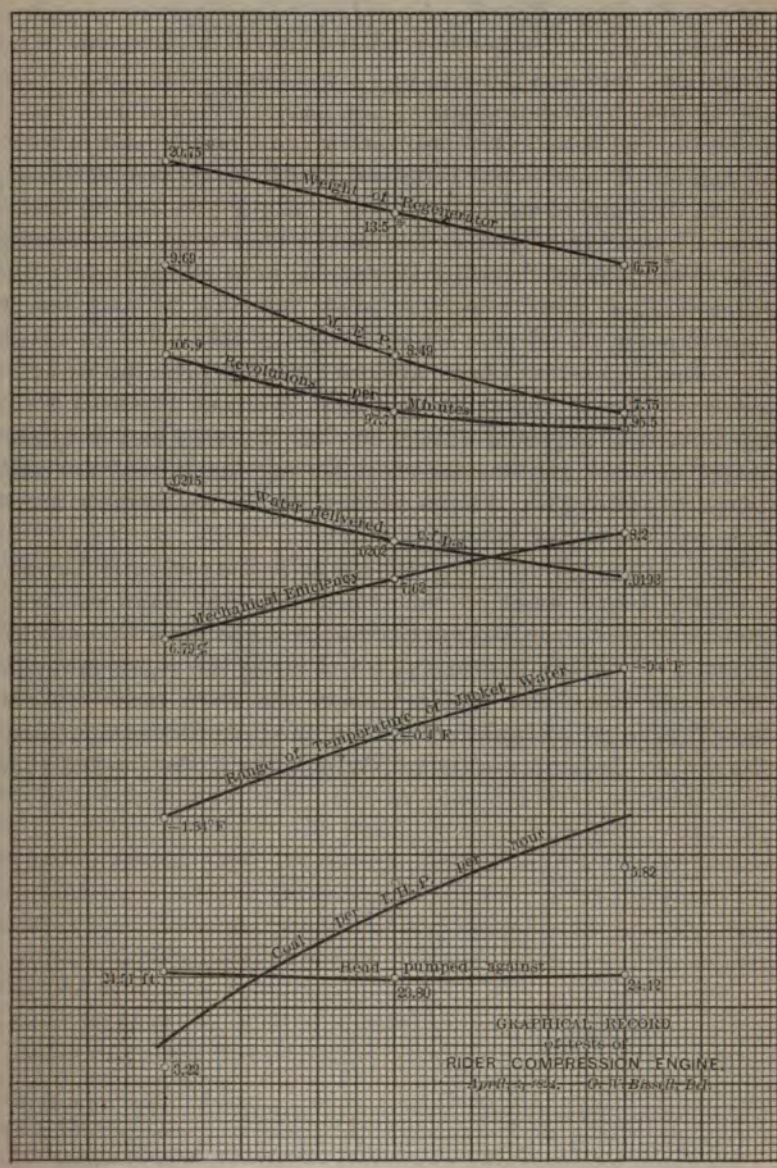
REMARKS.—The rise in temperature of the water from the pump to the weir was occasioned partly by the jacket-water from the working cylinder which was admitted to the discharge pipe after the final temperature of the main stream had been measured, and partly by the arrangement of the discharge pipe, which was carried through the upper part of the room in the vicinity of steam-pipes for a considerable distance before reaching the weir tank.

Inspection of the above data and results reveals the following:

Reducing the weight of the regenerator reduces the cooling effect of the engine upon the jacket-water, and finally results in actually raising its temperature;

And also increases the mechanical efficiency;

But at the expense of the duty and of the total efficiency as shown by the items under the heads respectively of "Water delivered, cubic feet per second," and "Coal per I. H. P. per hour."



DISCUSSION.

Mr. R. C. Carpenter.—The peculiar effect of the refrigerating cylinder of the Rider hot-air engine of the Sibley College Laboratory some two or three years ago is very well described in the paper by Professor Bissell. According to the experiments made at that time, the jacket-water surrendered heat to the air inside of the refrigerator, and was reduced to a lower temperature rather than take up heat from that source. The natural effect of this was to render the jacket-water not only useless but a positive detriment to the engine. I believe that Professor Bissell shows that on running without the jacket-water there was a slight loss rather than a gain in economy. This rather unexpected condition, reported to me at the time by Professor Bissell, led to an attempt on my part during the next year to find the reason for such peculiar results. Without going into details I will simply say that for some reason or another we have not been able to reproduce conditions which gave us results of the same character as those reported. The tests have been repeated a large number of times, and invariably the jacket-water was warmer when leaving the refrigerator than when approaching it. All the sections of the regenerator are in place, and the engine is in good order.

While I have not definitely decided as to the reason for reversal in results, I had come to the conclusion that there was some abnormal condition existing at the time that Professor Bissell made his tests. I will say that except for cleaning the regenerator, the engine has not been repaired since the tests made by Professor Bissell, yet it is quite certain that the jacket-water surrenders heat to the refrigerator in the normal manner at the present time.

DLXXXVIII.*

A NEW MECHANICAL FLUID.

BY CHARLES WALLACE HUNT, NEW YORK CITY.
(Member of the Society.)

A HYDRAULIC piston for adjusting journal boxes to take up the wear from use would have the desirable features of delicacy of adjustment and ample rigidity for any strain which it would be called upon to sustain. The cost of the accurate construction needed, and the probable leakage of the fluid, would usually prevent serious consideration of this method.

In canvassing the various methods of adjustment which could be used in the bearings of the connecting rods of our steam engines, the advantages and faults of this method were carefully investigated. The inevitable leakage of the liquid presented an almost unsurmountable barrier to its use. The thorough examination of its availability for this purpose, however, brought to mind the method of packing the plungers of mine-pumps used by Mr. Eckley B. Coxe, which has since been presented to this Society in a discussion of topical questions at the New York meeting, December, 1893.† This method was to fill the spaces for the packing material around the plunger with untried tallow, cut in pieces about an inch across, and kept under sufficient pressure to make it flow into the packing spaces. The tallow packing also lubricated the piston, and as it was gradually worn away, the membranes in the tallow were too large to pass between the plunger and its casing, and remained, forming under the pressure a mass of semi-fluid packing. If we consider this material as a fluid, we can conceive of it as a liquid with masses of animal membranes for atoms, instead of the usual conception of atoms of infinitesimal size.

After making this wide departure from our ordinary conceptions in molecular physics, we can take a further step by imagin-

* Presented at the Montreal meeting, June, 1894, of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† *Transactions American Society of Mechanical Engineers*, vol. xv., p. 599.

ing a liquid which was composed of atoms as large as bicycle bearing-balls. The anticipated trouble from the leakage of the

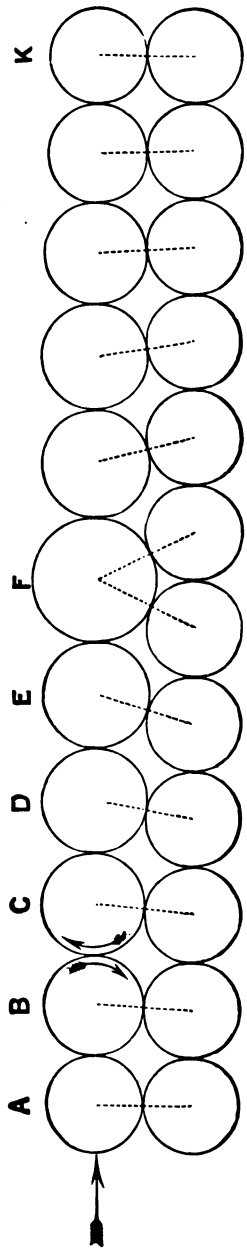


FIG. 227.

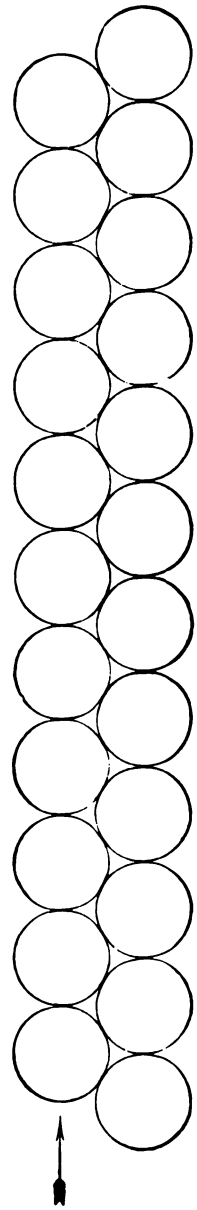


FIG. 228.

fluid would then disappear as a factor in the ordinary mechanical fitting in steam-engine work.

If the balls of this hypothetical fluid vary in size, their mutual pressures may be supposed to balance each other somewhat as shown in Fig. 227. The lower balls are of equal size, and the upper ones of varying diameter. The ball *A* is placed directly over the centre of the lower ball, and has no tendency to move in either direction, but the balls *B*, *C*, *D*, *E*, and *F* are each larger than the preceding ball, and each one rests upon the inclined surface of a supporting ball, all together throwing a horizontal pressure on the ball *F*. A similar series of balls is shown on the opposite of the ball *F*, and the whole system of balls from *A* to *K* is in equilibrium, with no tendency to move. The whole system, however, is in a position of unstable equilibrium, and will move if the lightest pressure is applied to the ball *A* in the direction of the arrow. If the balls were infinite in number, and the size limited, but with an infinite number of variations of diameter within the limits, the various local pressures would be balanced, and the mass would have the mobile property of a liquid.

The actual movement of the upper system of balls would result in a movement of revolution in each of the balls. Assuming that the friction of the lower ball would cause each one of the upper balls to revolve, the surfaces in contact in the upper system would move in opposite directions, as shown by the arrows in the balls *B* and *C*. This friction would depend upon the material, the smoothness of the surfaces, and the lubrication of the balls, and can be likened to that property of a liquid which is known as viscosity.

Under compression each ball of a mass of various sizes would have the resultant of all the pressures on it at its centre, and consequently would be in a position of equilibrium; but should the enclosing envelope change in form or capacity, a change in the direction of the pressures would take place; first, in the balls nearest to the movement, then in the adjoining ones, the balls severally moving in the direction of least resistance, until the resultant of the pressures again came in the centre of each ball.

If the mass of balls were all of one size they would interlock in a manner analogous to the illustration in Fig. 228, where a series of balls is shown, with a set of the same size resting upon

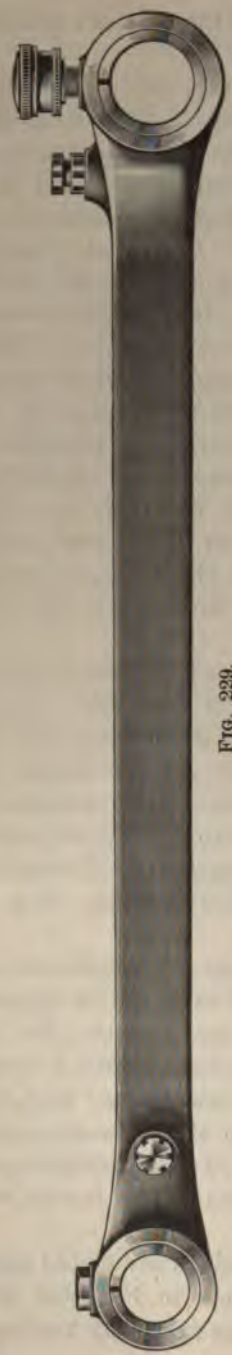


FIG. 229.

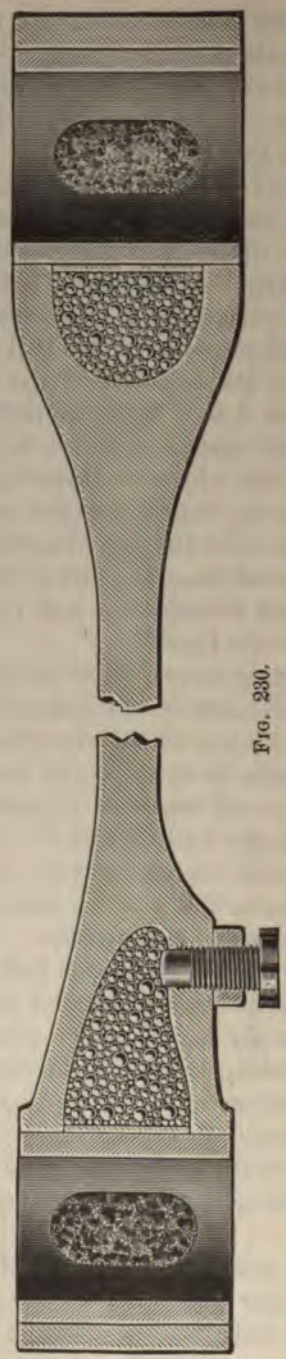


FIG. 230.

them, interlocked in the spaces of the lower set. If a pressure in the direction of the arrow is applied, it must raise each ball, and the pressure required would be in proportion to the distance, or to the number of balls moved. This rigidity resembles the property of a solid.

If these balls of equal size had different coefficients of expansion by heat, it could be imagined that at some higher temperature these balls would become of different sizes, when they would cease to interlock, the mass becoming a fluid above that temperature, a property analogous to the melting point of a solid.

If the balls had different coefficients of compressibility, a great pressure on the mass might change the relative diameter of the balls enough to change the solid mass to a fluid.

A mass of balls of various sizes, under pressure, does act like a liquid, as we have ascertained by testing in various ways. On the table, at the reading of the paper, is to be found a connecting rod of full size fitted up to be adjusted in this manner, just as it is used in practice, and also a model with a glass front and a spring piston, which permits the individual motion of the steel balls to be seen when the adjusting screw is turned. The mass is kept mobile by putting sufficient pressure on the movable piston.

Fig. 229 is an elevation, and Fig. 230 is a cross-section of a solid-end connecting rod, having an annular brass bushing with an opening, in order that adjustment for wear can be made. One side of the bushing is held in place and adjusted by the pressure of a mass of hardened steel balls, varying in diameter from one-eighth to one-quarter of an inch. Between the bushing and the balls is a thin plate of hardened steel to prevent the balls from indenting the softer metal. The adjustment for wear is made by forcing the mass of balls forward with a hardened steel adjusting screw. (Fig. 231.)

When the wear has become so great that the range of the adjusting screw has been exhausted, a few more balls are inserted under the point of the screw, and an adjustment of the bearing made as before. On the crank end of the rod, the adjusting screw is on top, but on the cross-head end it is on the front side, clear of the cross-head, where the adjustment is as conveniently made as it is on the crank end.

We have been running last year forty bearings of this type in various places, in the East and in the West, work-

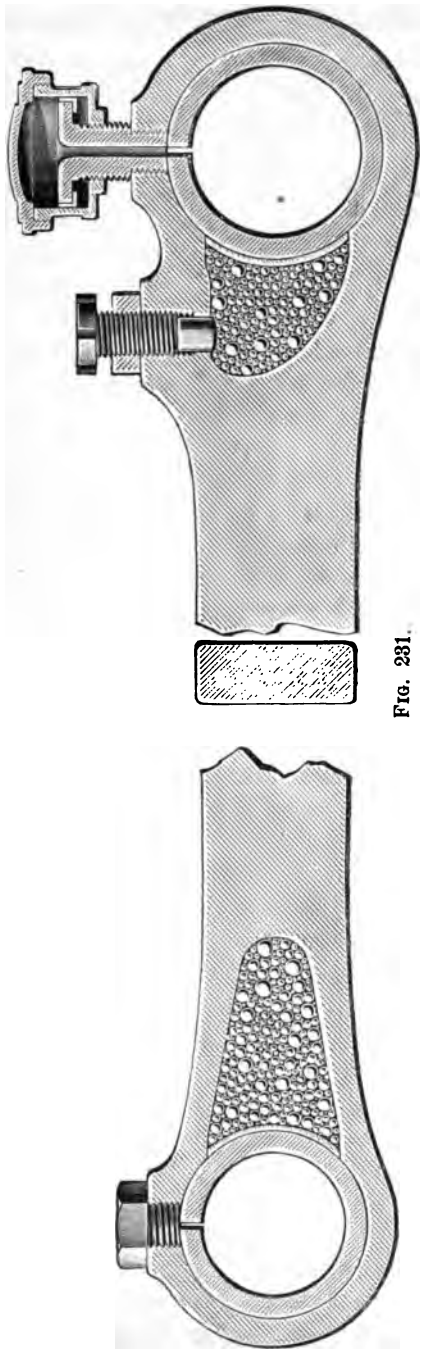


FIG. 231.

ing with steam-boiler pressures of from 80 to 160 lbs., giving pressures per square inch of projected area of the pins ranging from 600 to 1,000 lbs. The pressure on the balls per square inch is about 50% greater. If we assume that the balls are about one-eighth of an inch in diameter, and that 60 balls are in contact with the bushing on each square inch, the pressure sustained by each ball would be from 15 to 25 lbs. Assuming that the elastic limit of the steel in the connecting rod shown is 30,500 lbs. per square inch, and that the hardened steel balls have a pressure of 25 lbs. on each one, we would expect that each ball would imbed itself in the steel envelope until the area sustaining the pressure was strained to the elastic limit. This area would be $\frac{1}{1270}$ of a square inch for each ball, which is almost exactly equal to the area of a circle $\frac{1}{100}$ of an inch in diameter. At first the bearings require frequent adjustment, but the balls soon get in a fixed position, and then they need adjustment for the wear of the bushing only. The adjustment of these bearings is apparently as delicate and positive as though a liquid was used. In cases where the adjusting

screw was purposely made to turn freely, the adjustment could easily be made by the thumb and fingers, and if care was not used too great a pressure on the pin would result.

The connecting rod for a 10-inch diameter cylinder steam engine here shown is a steel casting made by I. G. Johnson & Co., fitted with an adjusting screw having fourteen threads to an inch, and a pressure area of balls on the bushing eighteen times the area of the screw. If the screw is turned with a force of 25 lbs. on an 8-inch wrench, the friction of the screw absorbing 25% of the force, the pressure on the bushing would be over 50 tons. While this pressure might be used in other applications of this novel fluid that have been proposed, it was clearly out of place in connecting-rod bearings, and we now make the head of the adjusting screw of such a form that an ordinary wrench cannot be used, and the adjustment made only by a spanner which will, by its peculiarity, remind the engineer that care is necessary in the adjustment.

DISCUSSION.

Mr. George I. Rockwood.—I would like to ask the author of the paper how he thinks this method of adjustment would apply to large engines?

Mr. Hunt.—Whether this device would act more favorably on large bearings than on small ones I do not know. It was a new thing, and we put it on some engines to try it and see how it would work. It has worked very nicely in the way of being fluid. By moving the adjusting screw of the model backwards and forwards to adjust the bearing you will see the balls adjust themselves into position, and frequently see the whole of the balls move around in a circle something like a whirlpool. I see no reason, however, why for large bearings it would not be especially efficient. The larger the bearings the more efficient it should be, because the balls would be relatively smaller.

Mr. W. S. Rogers.—While discussing the question of balls for packing and adjusting purposes, I have used bird shot for the past twenty years, at various times, for packing stuffing boxes of engines and steam-pumps, and don't know of a better thing on earth. I have also used it on the piston rods of both air and steam cylinders, of air-brake pumps on locomotives, and it is used by many others with best results. I never heard it

called a mechanical fluid before, but think Mr. Hunt has rightly named it.

Mr. Rockwood.—For my part, it appears to me that not only is this adjustment a thoroughly novel thing, but that the limit of its adaptability is not really hinted at by the application that we have here; for it must be much cheaper and handier than an ordinary strap, gib, and key connection, and should find a wide application in valve-rods and connecting rods such as are now provided with non-adjustible bushings, a style made so prevalent by competition in, for instance, pumping machinery. Those of us who have watched the operation of a modern high-duty, direct-acting pump have, no doubt, often been tempted to step up and wriggle the valve-gear connections with a view to learning the limit of lost motion in the bushings deemed allowable before the connection is condemned. It is often a wonder, indeed, that the valve-gear can have an appearance so drunken and yet keep up the regular operation of the pumps. It seems to me here is a good place for the application of this stub-end, and possibly it is applicable to much heavier bearings. I look to see this invention have a wide application.

Mr. John A. Brashear.—It may be interesting to recall that Dr. H. Draper made experiments, in 1863 and 1864, on the flow of small balls through an orifice, and found their action was exactly like a liquid except this: Say you fill a cylinder ten feet high, the flow is exactly the same when it is nearly empty as when it is full, and it was found so efficient for obtaining smooth and regular movements in celestial photography, that he used such a driving mechanism for quite a long while very successfully. The balls flowed through an orifice exactly the same as a liquid, except that the rate was not affected by change in the height of the column.

Mr. Hunt.—I would like to say in reference to the peculiarity mentioned by Mr. Brashear, which is very interesting, that it is probably caused by the friction of the balls under pressure against each other and against the envelope from which they flow. The friction of the balls against each other is quite a material amount, and it is spoken of in the paper as similar to viscosity. I think that the lack of increased flow by increased height was caused by the friction of the balls on each other and on the orifice from which they flowed.

DLXXXIX.*

CYLINDER PROPORTIONS FOR COMPOUND ENGINES DETERMINED BY THEIR FREE EXPANSION LOSSES.

BY FRANK H. BALL, NEW YORK CITY.

(Member of the Society.)

AT the Chicago meeting of this Society, forming part of the International Engineering Congress, a paper was presented entitled "Compression as a Factor in Steam-Engine Economy," † in which a theory was elaborated for measuring and harmonizing the free expansion losses at both ends of the diagram. In the paper referred to it was also suggested that this system of measurement might furnish valuable information as to the relative losses from free expansion in the several cylinders of compound engines, and the Society was promised a paper on this subject at a future meeting.

In offering this paper as a fulfillment of that promise, the author is aware that he is widening the application of a law suggested in the former paper, which law was not as generally accepted as had been anticipated. This scepticism on the part of some of our leading members fortunately led to a series of experiments, since conducted at the Stevens Institute of Technology, which experiments are the subject of a paper presented at this meeting of the Society by Prof. Jacobus who conducted the experimental work. The results as reported seem to confirm the law in question, and it is therefore with greater confidence that its further application is here made.

To make this application, let it be assumed that cylinders are to be selected for a triple expansion engine, where the boiler pressure is 150 lbs. above atmosphere, and the vacuum gauge shows 26 inches. Allowing for wire drawing, let it be assumed that the initial pressure in high cylinder will be 160 lbs. abso-

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* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† Volume XIV., *Transactions American Society Mechanical Engineers*, page 1067.

lute, and the back pressure in low cylinder 3 lbs. absolute. The total range of pressure is therefore 157 lbs., and the corresponding range of temperature 221° Fahr.

The object of compound engines being to reduce cylinder condensation by dividing the range of temperature judiciously between two or more cylinders, the first step is to decide through what range of temperature each cylinder shall work. In doing this, the desirability of a tolerably uniform division of work beween the various cylinders forming the system must not be overlooked, although it cannot be considered good engineering to impair the economy of the engine materially to accomplish this result, as each engine of the system may be built to carry any load found desirable to put upon it. If the range of

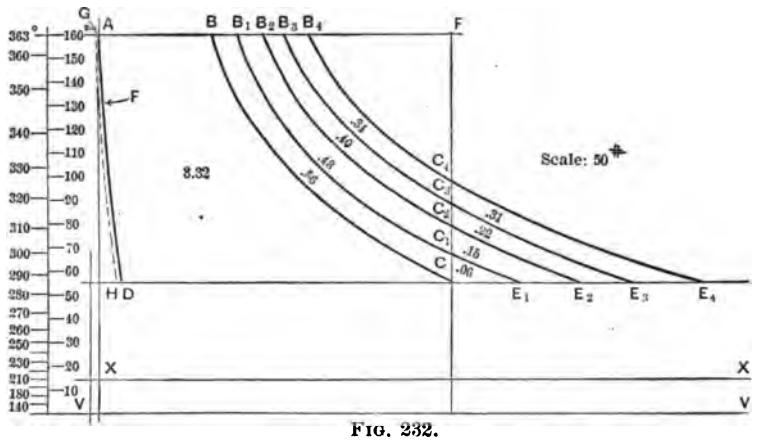

FIG. 232.

temperature is divided equally between the three cylinders in the proposed engine, the greater internal surface of the low cylinder would warrant the expectation of greater cylinder condensation than in the smaller cylinder, and if so, the total condensation can be reduced by giving the low-pressure cylinder less range of temperature and the high cylinder more.

Cylinder condensation in this investigation, whether considered relatively or collectively, must be made to include the steam consumed in the jackets of each cylinder, if jackets are used. Whatever may be true in regard to the best range of temperature for each cylinder, the logic of what is to follow will apply with equal force, and therefore, for the purpose of illustration, it will be assumed that the temperature is to be equally

divided. This will require that the high cylinder works between 160 lbs. absolute and 57 lbs., the intermediate between 57 and 16 lbs., and the low cylinder between 16 lbs. and 3 lbs. absolute pressure.

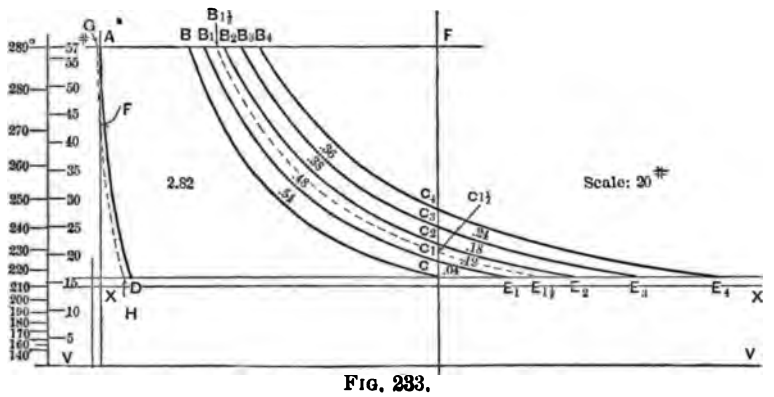

FIG. 233.

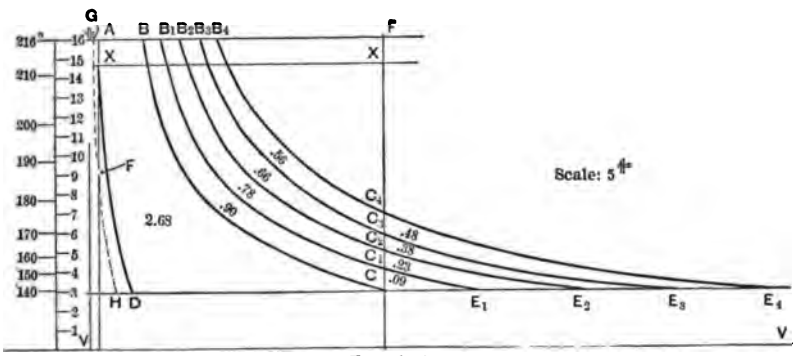

FIG. 234.

Figs. 232, 233, 234 and represent theoretic diagrams between each of the three divisions of pressure mentioned. In each case "*V V*" represents the vacuum line, and "*X X*" the atmospheric line, and in each case "*A F*" represents the piston travel. The clearance is assumed to be 4% of the piston displacement, as indicated. For convenience in constructing the curves and measuring their enclosed areas, the following scales have been chosen :

Fig. 232	Scale 50 lbs. to the inch
Fig. 233	" 20 " " " "
Fig. 234	" 5 " " " "

In each case "*A B C D*" represents a theoretically perfect diagram, so far as free expansion is concerned, because expansion is carried to the line of return pressure, and compression fills the clearance space to initial pressure. It is hardly necessary to say that the curves here shown are Mariotte curves, and not Adiabatic, as the latter are seldom used and are considered an unnecessary refinement in this investigation. The successive curves in each diagram, which follow the curves "*B C*," represent later points of cut-off, and they are continued beyond the limits of piston travel until they intersect the line of return pressure. The areas of enclosed spaces are indicated by figures. Thus in Fig. 234, the area of the theoretically perfect diagram "*A B C D*" is 2.68 inches, and "$B\ B_1\ C_1\ C\ B$" is .90 inches, etc.*

The next step is to determine the best point of cut-off for each cylinder. In this investigation each cylinder must be considered separately, and treated as though it was a single cylinder engine working between the limits of pressure indicated, and it may be asserted without fear of successful contradiction that if any cylinder of a compound engine is not realizing the highest economy obtainable from a single cylinder engine working between its limiting pressures, then the engine as a whole is falling short of its possibilities. It is also true that if because of cylinder condensation it is not economy to expand to the line of back pressure in a single cylinder engine, the same is true of every cylinder of a compound engine, it being only a question of the degree of free expansion permissible in each case.

To those who believe that there ought to be no "drop" in any of the cylinders of a compound engine except the low, the foregoing will seem rank heresy. They argue that if there is "drop" in the high cylinder there is free expansion waste, and by earlier cut-off in the low cylinder the receiver pressure may be raised until the drop in high cylinder disappears, thus eliminating free expansion and improving the economy of the system. This is a plausible fallacy which represents only one side of the question, the other side being that by raising the receiver pressure the range of temperature is increased in the large cylinder, thereby increasing the cylinder condensation in this cylinder without effecting a corresponding reduction of condensation in

* For convenience of publication, the diagrams of Figs. 232, 233, and 234 have been reduced in size, and therefore, while the areas remain relatively the same, the figures given are the actual areas of the original diagrams.

the smaller cylinder. Looking at this question from another point of view, let it be admitted for the moment that the economy of the engine will be improved by raising the receiver pressure until the "drop" in high cylinder disappears. Then considering the high cylinder alone, we have a diagram in which expansion is carried to the line of back pressure, which cannot be considered the most economical diagram from any engine whose internal condensation is not in proportion to the steam used, which is the recognized condition of all steam engine cylinders. Therefore the economy would be improved by using a cylinder of smaller diameter, with less exposed surface for condensation and necessarily some "drop" at exhaust opening. Then if it is true that better economy will be realized by raising the receiver pressure again to eliminate the drop as before, it only requires a few successive stages of this development to dispense with the high cylinder altogether, completing the whole expansion in the low cylinder, and at the same time improving the economy at every step in that direction, reaching the highest economy when the low cylinder covers the entire range of expansion and the engine becomes a single cylinder engine instead of a compound. To those who still adhere to the belief that what is true of one cylinder of a compound engine is not true in any degree of the others, and that one cylinder of an engine may be wasteful in its preformance, without affecting the aggregate performance of the system, the reasoning of the following pages will not be convincing. To those who believe that each cylinder must realize the most economical performance for a single cylinder working between its limits of pressure, and believe also that cylinder condensation makes it impossible to obtain this by expanding to the line of back pressure in any cylinder, the method of investigation which is to follow will be of interest.

Referring again to Figs. 232, 233, and 234, let us first investigate the low-pressure cylinder represented in Fig. 234 as being in some respects the most important one of the system. If 3 lbs. absolute be assumed to be the back pressure in this cylinder, the diagram "*A B C D*" represents full and complete expansion, and will correspond with the highest economy of this cylinder if there was no condensation, or if cylinder condensation was a uniform per cent. of the steam used. This diagram, carrying expansion to 3 lbs. absolute, represents a total of more than 53

expansions in the system, which is recognized as being far beyond the economical limit. Following the successive expansion curves from "B_1 B_2 B_3 B_4," it will be seen that "B_1" adds an area to the useful diagram of .9 inches, or more than $\frac{1}{3}$ of the area of "A B C D," and the free expansion loss occasioned by this curve is only 10% of the useful area that has been realized. This loss is represented by the area of "C_1 E_1 C." Substituting curve "B_2 C_2" for "B_1 C_1," the further addition of useful area is only .78 inches, which is accompanied by an additional free expansion loss of 29% of this amount, and in the same manner "B_3 C_3" only adds .66 inches of area with a loss of 57%, and finally "B_4 C_4" with a useful area of only .56 inches entails a loss of 85% of this amount in consequent additional free expansion. As between "B_3" and "B_4" then, it is not likely that an amount of cylinder condensation is ever found in practice that would make it economy to add this last area to the diagram with its enormous free expansion loss; and the terminal pressure of "C_4," representing 22 expansions in the system, is, therefore, probably too high, and we must look for the best results between 22 expansions corresponding to "C_4" and 53 expansions due to terminal "C."

If the exact amount of cylinder condensation due to each of the five points of cut-off was known, the indicated areas of the diagrams would furnish the data necessary to determine at once the exact point of cut-off where the highest economy would be realized, and just to the extent that we are able to correctly estimate this cylinder condensation, will we be able to determine correctly the best number of expansions in each cylinder and in the system. It must be here understood that it is not intended in this paper to determine absolutely the best cylinder ratios or the best number of expansions for the conditions that have been assumed because of the lack of exact data regarding cylinder condensation, but it is the purpose of the author to show that by this method of investigation even an approximation of the cylinder condensation enables the engineer to decide the question of cylinders with a comparatively small limit of possible error, and this error may be reduced toward zero just in the proportion that the truth regarding cylinder condensation is known. To illustrate the manner of applying this knowledge with the use of the diagrams of Figs. 232, 233, and 234, various amounts of cylinder condensation will be assumed and the problem

worked out for each case. Before doing this it may be interesting to note the relatively greater wastefulness of free expansion in the low-pressure cylinder than in the others. Referring to Fig. 234, the removal of the point of cut-off from "B" to "B_4," adds to the useful diagram the area of "$B\ B_4\ C_4\ C\ B$" equal to 2.90 inches. The free expansion of 4 lbs. at "C_4" results in a loss of useful work represented by the area of "$C_4\ E_4\ C$" equal to 1.18 inches, or about 41% of the former. Referring to Fig. 232, representing the high-pressure cylinder, and repeating the same calculation, the area of useful work of "$B\ B_4\ C_4\ C\ B$" equals 1.78 inches, and the drop at "C" of 40 lbs. causes a loss represented by "$C_4\ E_4\ C$" equal to .78 inches area, or about 41% of the useful work in this case also, so that by comparing the two we see that in both cases the useful areas of "$B\ B_4\ C_4\ C\ B$" are accompanied by a free expansion loss of about 41%, but in the low-pressure cylinder the terminal drop is only 4 lbs. as against 40 lbs. in the high cylinder.

The foregoing ought to furnish food for thought to the engineer who is chiefly concerned about preventing drop in the high and intermediate cylinders. Referring again to Fig. 234, and resuming the consideration of cylinder condensation, let us assume certain quantities to represent this cylinder condensation, and thus illustrate the manner of applying more exact knowledge on this subject. It is evident that the conditions which make late cut-off desirable are large condensation and a constant quantity at every point of cut-off, and the reverse conditions, viz: small condensation, varying for each point of cut-off, would call for early cut-off. In illustrating this subject two rather extreme conditions have been chosen; one where the cylinder condensation is assumed to be 25% of the steam accounted for by the indicator at latest point of cut-off, and this amount to remain undiminished at the earlier points of cut-off, and the other condition where cylinder condensation is assumed to be only 15% instead of 25% as above, and to decrease at each of the earlier points of cut-off in the following manner:

14%	when	cutting	off at	B_3
13%	"	"	" "	B_2
12%	"	"	" "	B_1
11%	"	"	" "	B

Fig. 235 gives a graphic illustration of the effect on the econ-

omy of the low cylinder produced by the two assumed conditions as to cylinder condensation. Referring to Fig. 234, the steam accounted for by the indicator at latest point of cut-off may be represented by the area of "$A\ B_4\ E_4\ D\ A$," which equals 6.78 square inches. If the steam condensed on entering the cylinder is 25% of that accounted for by the indicator at latest point of cut-off it may be represented by 25% of the area 6.78″ = 1.69″.

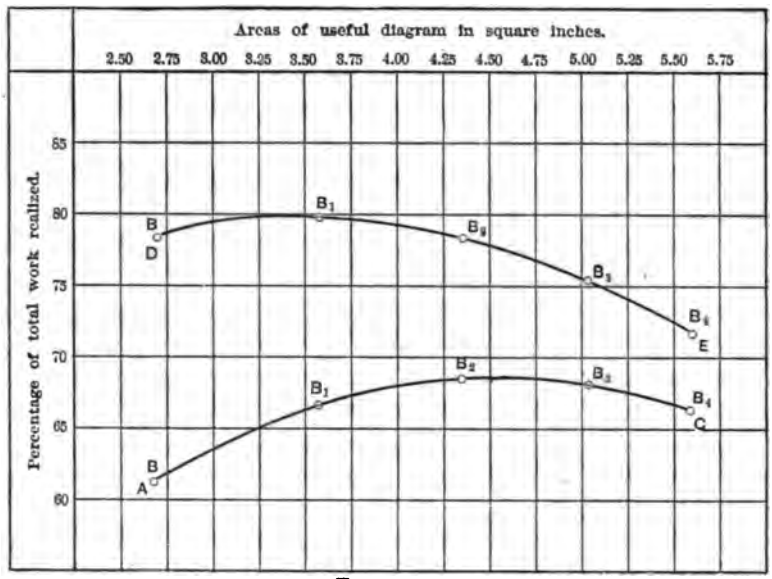

FIG. 235.

This last area then, under the assumed conditions, will be a constant loss at every point of cut-off. An additional loss by free expansion occurs at every point of cut-off later than "B."

These free expansion losses are represented by the indicated areas beyond the limits of piston travel, which combined with the area representing condensation, and compared with the areas of useful diagram, produce the curve "$A\ C$" (Fig. 235) in the following manner. Measurements on abscissa correspond to the useful areas of diagrams at the five points of cut-off indicated. Measurements on ordinates are obtained as follows:*

For cut-off "B" the area of useful work is 2.68 inches, and the assumed constant loss by condensation of 1.69 inches makes

* This method is fully illustrated on page No. 1067, Vol. XIV., *Transactions American Society Mechanical Engineers.*

the total area due to the steam consumed equal to 2.68 inches +, 1.69 inches = 4.37 inches. Of this amount the area actually realized, 2.68 inches, is 61.3%, which is measured on ordinates and establishes the point "*B*." For cut-off "B_1," in the same manner as before, we have useful area of

"A B_1 C_1 C D A"	3.58 inches
Free expansion of "C_1 E_1 C"	.09 "
Assumed condensation	1.69 "
Total area due to steam used at cut-off "B_1"	5.36 inches

Of this area only 3.56 inches has been realized, or 66.7%, which establishes point "B_1" on curve "*A C*" (Fig. 235). The succeeding points on this curve are found by the same process, using the indicated areas at each progressive step.

Curve "*D E*" is produced by the same method, the only difference being that the smaller and varying condensation is assumed; thus, instead of the constant area of 1.69 as assumed in curve "*A C*," the condensation is supposed to be represented by the following areas:

.74	inches	for	cut-off	"B"
.81	"	"	"	"B_1"
.88	"	"	"	"B_2"
.95	"	"	"	"B_3"
1.02	"	"	"	"B_4"

These quantities are substituted for the constant quantity, 1.69 inches, used with curve "*A C*" and the result is curve "*D E*."

From an examination of these curves it appears that with the conditions assumed for curve "*A C*" the highest economy is obtained at or near cut-off "B_2," while with the conditions which produce curve "*D E*," the highest economy is found at or near cut-off "B_1."

If the condensation assumed for either of these curves was known to be correct, then the best point of cut-off would at once appear; but even without exact knowledge as to condensation, the range covered between the two assumed conditions is so great that the actual condensation would probably produce a curve between the two, and if so, then the terminal pressure at release in low-pressure cylinder of the proposed engine should not be less than 4 lbs. absolute, nor more than

5 lbs., and the corresponding number of expansions of the system will not be more than 40 nor less than 32. For the purpose of keeping the low-pressure cylinder as small as possible, the fewest number of expansions should be employed that promises approximately the best economy, and therefore we will select 32 expansions, and the expansion curve in low-pressure cylinder will be "B_2 C_2."

Referring to Fig. 235, the lower curve assumes the greatest cylinder condensation, and on this assumption "B_2" appears to be the best point of cut-off. The condensation here assumed,

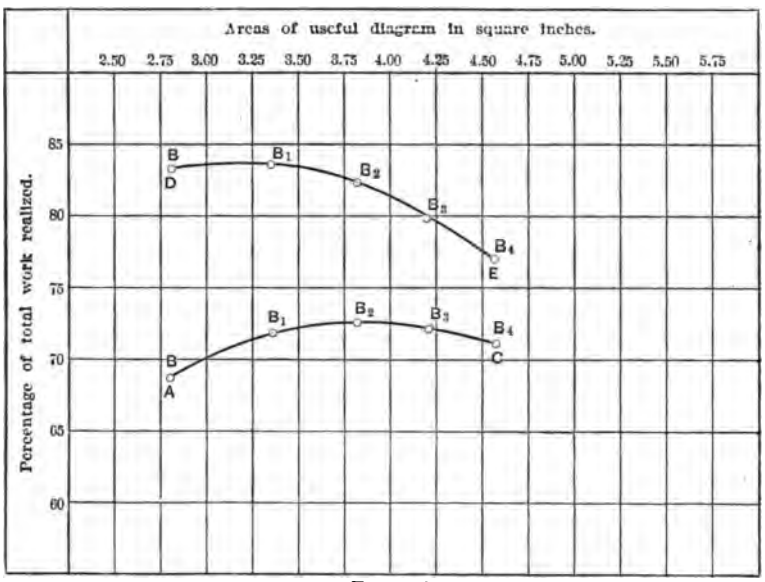

FIG. 236.

if referred to the area of "A B_2 C_2 C D" will be found to equal 47% of the steam accounted for by the indicator when cutting off at "B_2," which, in view of the cut-off being later than 1/4 stroke and the total range of pressure in the cylinder only 13 lbs., is an altogether improbable amount. It is also equally improbable that the actual loss by condensation would not decrease slightly with earlier points of cut-off, so that in selecting 32 expansions for the proposed engine, it is done with the idea of using the least permissible number that will even approximate the best economy. Having determined that 32 expansions will be obtained in the proposed engine, and that "A B_2 C_2 C D" of

Fig. 234 will be the diagram of low-pressure cylinder, we will next proceed to investigate the intermediate and high-pressure cylinders.

Beginning with the intermediate, Fig. 233 represents a series of possible diagrams between the pressures that have been allotted to this cylinder, and Fig. 236 represents the economy of each of these points of cut-off under the two extreme conditions of condensation that were assumed in Fig. 235. The method of locating the points on the curves of Fig. 236 is exactly the same as that of Fig. 235, merely substituting the areas of Fig. 233 for those of Fig. 234, and need not be again explained. Following the curves of Fig. 236, it appears that for the smallest condensation the best economy is at or near cut-off "B_1," while with the largest condensation the best result is with cut-off somewhat later than "B_2." Between these two points, then, we must probably look for the desired point of cut off, and, as before stated, if the exact condensation for each point was known it could be very quickly determined. As between this cylinder and the low we may assume that the condensation will be somewhat less in the smaller cylinder because of its smaller area of surface. This would be favorable to earlier cut-off, and the practical limitations as to size of cylinder do not interfere, as is the case of the low cylinder. On the other hand, free expansion is not a total loss in either the high or intermediate cylinder, as its superheating effect reëvaporates a certain quantity of the moisture in the steam, thus delivering to the receiver an appreciably greater volume of steam than that accounted for by the indicator at exhaust opening if much "drop" occurs. For this reason, "drop" is less objectionable in these cylinders than in the low, where no such redeeming feature is found. After due consideration of these modifying influences, it is not improbable that about midway between "B_1" and "B_2" will approximate the best point of cut-off for this cylinder, and to continue the illustration of the proposed method, the dotted curve "$B_{1\frac{1}{2}}$ $C_{1\frac{1}{2}}$" will be selected as the desired curve.

Proceeding in the same manner with Fig. 232, representing the high-pressure cylinder, and what has just been said about the intermediate cylinder applies to this equally as well. From a study of the curves of Fig. 237, as representing the performance of the high-pressure cylinder at the various points of cut-off under the same conditions as to condensation that were

assumed in Figs. 235 and 236, and keeping in mind that unlike the low cylinder no practical difficulty exists as to its size, we may with confidence select "B_1" as a point of cut-off that promises approximately the best results that may be obtained from this cylinder.

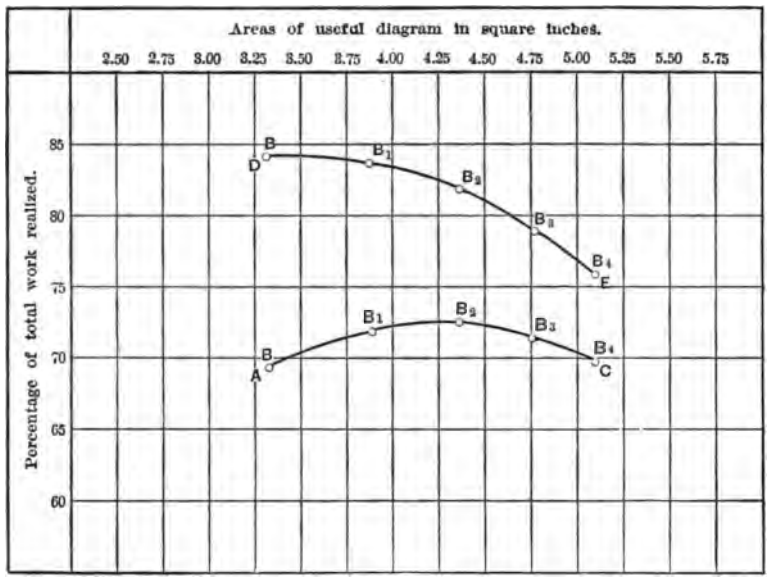

FIG. 237.

We have now established the expansion curves that we desire to produce in each cylinder, and to prevent confusion Figs. 238 239, and 240 represent diagrams from the three cylinders, in which only the desired expansion curves appear.

So far, in this investigation, no attention has been paid to the compression curves further than to state at the beginning that the compression curves shown are full compression curves which entirely fill the clearance spaces by compression, and rise without interruption to the initial pressure of the diagrams. Under no condition can these curves be the most economical, unless it has been shown that the best economy is obtained by expanding fully to the line of back pressure, as "$B\ C$" (Figs. 232, 233, and 234). Following the law relating to compression curves, suggested in Vol. XIV. of the *Transactions of the American Society of Mechanical Engineers*, page 1070, the curves $H\ F$ have been produced as the compression curves in each case that

correspond with the expansion curves that have been adopted. The completed diagrams, then, as sought to be produced, are represented by full lines in Figs. 238, 239, and 240. The next step is to ascertain the ratio of cylinders which will produce these respective diagrams, and to do this the diagrams must be compared as to the relative volumes of steam which they indicate. A very convenient graphical method for doing this is the following:*

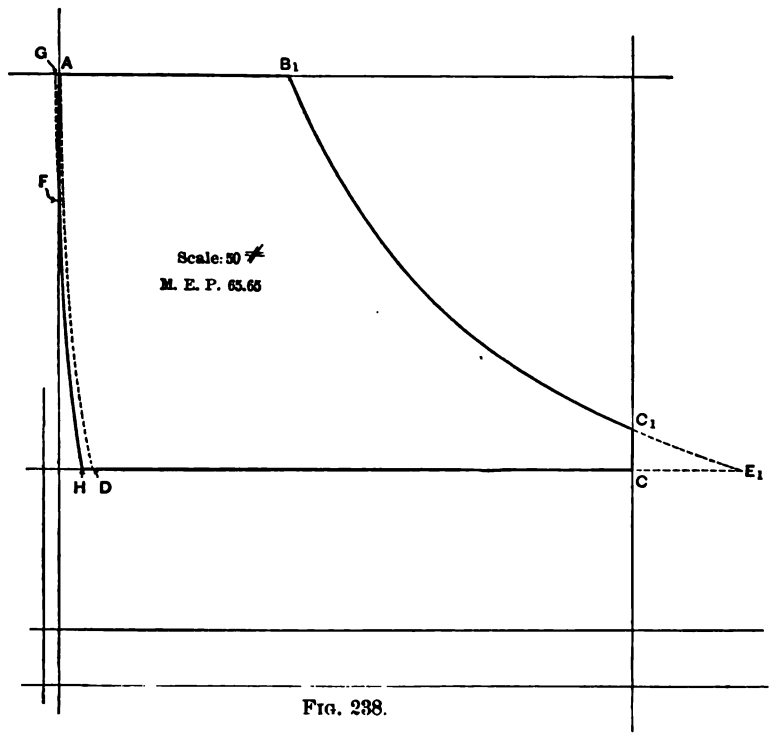

FIG. 238.

First, continue the compression curves "*H F*" of Figs. 239 and 240 to the line of highest pressure of each diagram, as shown.

Next continue the expansion curves of Figs. 238 and 239 to the intersection of the line of lowest pressure of each diagram, as shown.

Assuming that the lengths of these diagrams representing the piston travel are the same, and that the line "$G\ B_2$" in Fig. 240

* The author is indebted to his son, B. C. Ball, member of the class of '95 at Stevens Institute of Technology, for this method, which is believed to have been original with him.—F. H. B.

represents the same pressure as the line "$H\ E_{1\frac{1}{2}}$" (Fig. 239), it is only necessary to compare the length of the line "$G\ B_2$" with "$H\ E_{1\frac{1}{2}}$," and the inverse ratio will be the ratio sought. Thus, in this case, the line "$G\ B_2$" measured, with a scale of 100 to the inch, measures 92, and in the same manner "$H\ E_{1\frac{1}{2}}$" measures 368, therefore the ratio of the intermediate to low cylinder will be $\frac{368}{92} = 4$.

By the same method "$G\ B_1$" (Fig. 239) measures 104, and

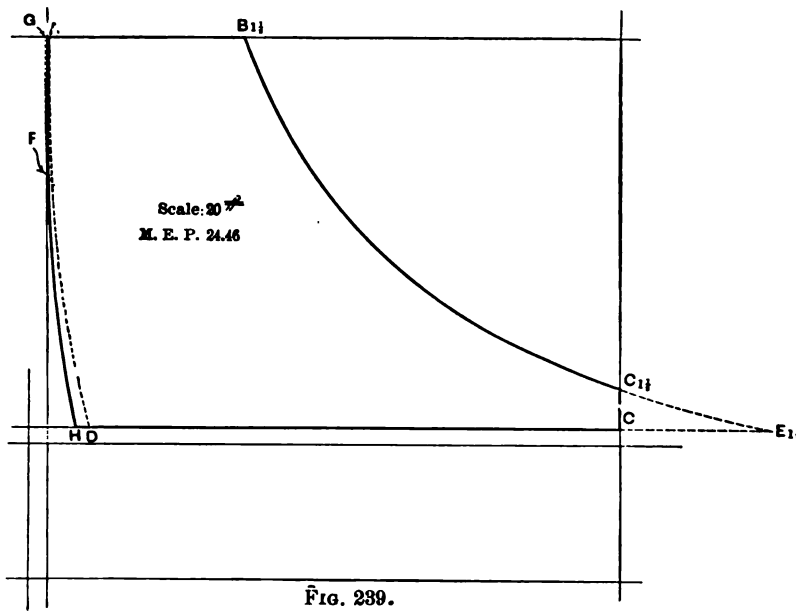

FIG. 239.

"$H\ E_1$" (Fig. 238) = 342, and therefore the ratio of high to intermediate cylinders will be $\frac{343}{104} = 3.3$.

Reviewing these figures we have the ratio of high to intermediate 3.3, and of intermediate to low 4, and consequently of high to low 13.2.

It sometimes may be more convenient in measuring cylinder ratios by the method just described to extend both the compression and expansion curves of the lower diagram upward to the line of terminal pressure of the next higher diagram, and use this line for measuring the ratios. The result will be the same in either case.

If these ratios are used, and the valves of each cylinder are set to give the steam distribution indicated, the actual diagrams from the engine will approximate very closely to those of Figs. 238, 239, and 240, except as they may be slightly modified by

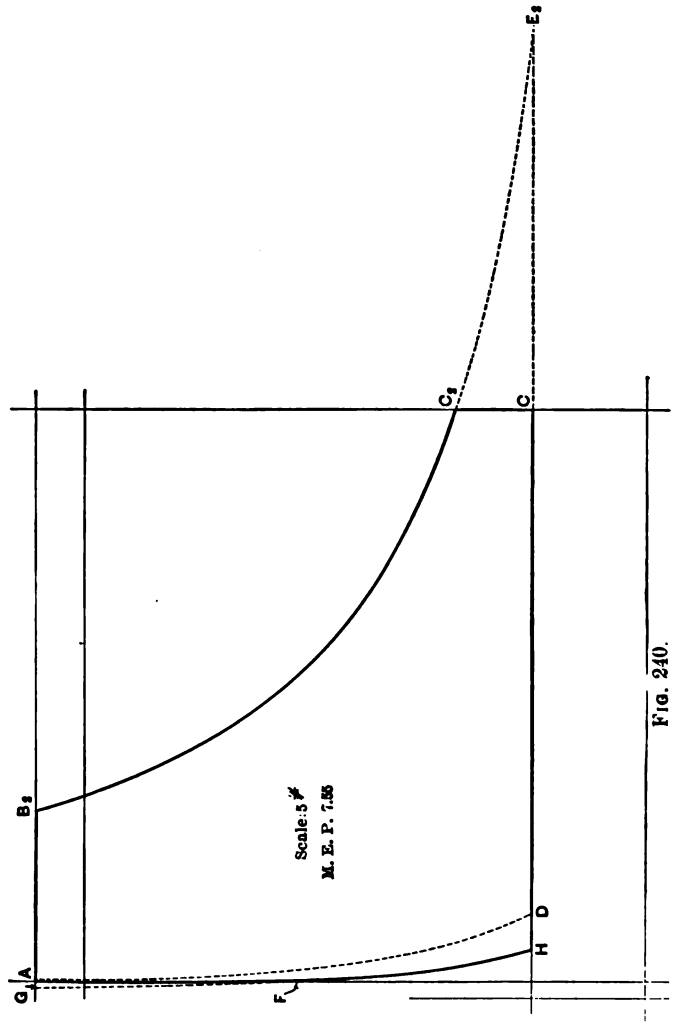

Fig. 240.

cylinder condensation. Where steam-jackets are used, the loss by cylinder condensation after cut-off in high-pressure cylinder is largely restored by heat from the jackets, so that frequently no allowance need be made in the ratios of cylinders for this loss.

With unjacketed cylinders a progressive deficiency will appear in each successive diagram of the system as compared with the theoretical diagrams, unless an allowance is made in the cylinder ratios to compensate for the progressive loss occasioned by condensation in the cylinders.

Continuing the study of the diagrams of Figs. 238, 239, and 240, the same diagrams appear in Fig. 241, reduced to the scale of the high-pressure diagram, and in Fig. 242 they are reduced

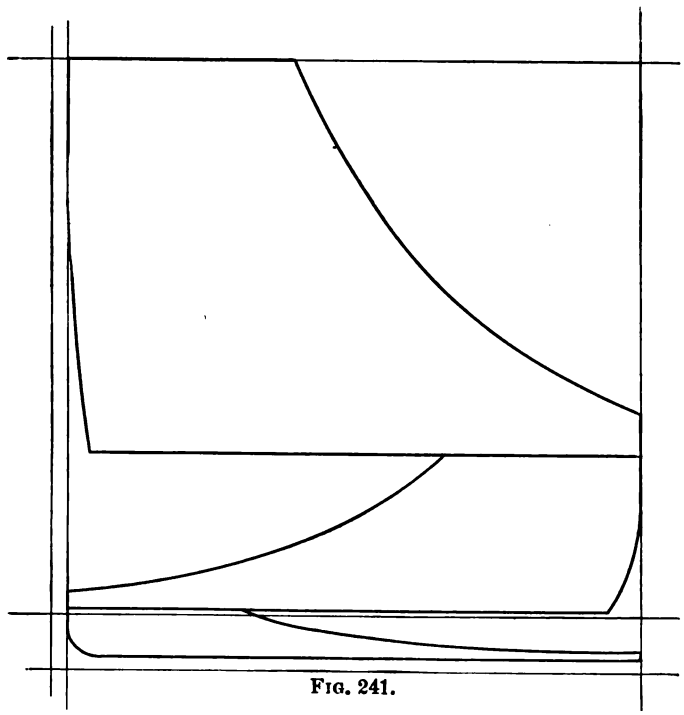

FIG. 241.

in length to correspond with the respective cylinder ratios, thus representing the total expansion referred to the low-pressure cylinder.

The mean effective pressures of these diagrams are as follows:

High cylinder	65.65 lbs.
Intermediate cylinder	24.46 lbs.
Low cylinder	7.55 lbs.

Or, referred to the low cylinder, as follows:

High cylinder	4.96 lbs.
Intermediate cylinder	6.10 lbs.
Low cylinder	7.55 lbs.
Total	18.61 lbs.

These figures show a progressive increase of indicated work in each cylinder from the high to the low.

A more even division of work would be obtained by decreasing the range of pressure in the low cylinder, and increasing the

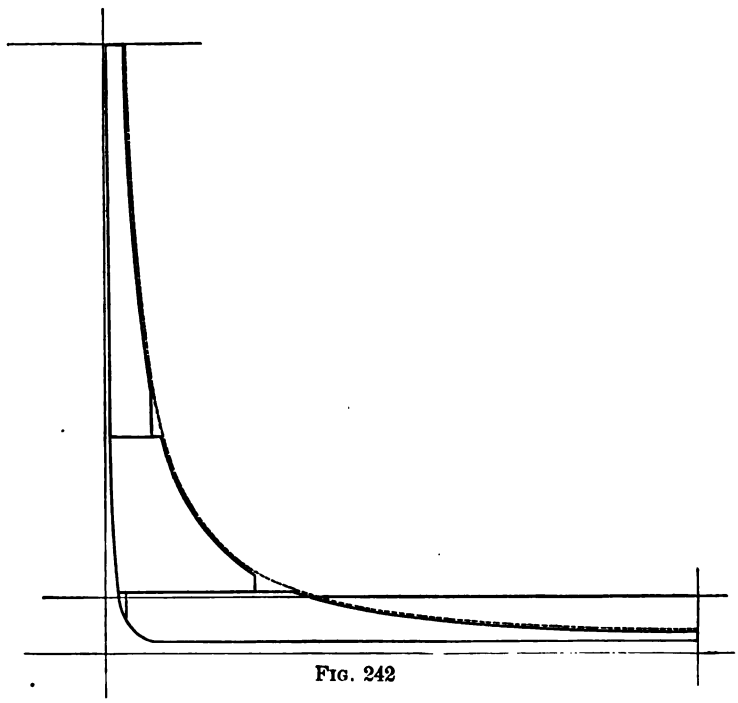

FIG. 242

range in the high. It has already been suggested, that because of the relatively larger area of surface of the low cylinder, a modification of this kind would probably reduce the total condensation; therefore it seems desirable that such a modification should be made, but the work will not here be reviewed for that purpose.

GENERAL CONCLUSIONS.

First, that in current engineering too few expansions are obtained in compound engines for best economy.

Second, that with 150 lbs. pressure and a good vacuum at least 32 expansions should be realized in a triple-expansion engine.

Third, that the cylinder ratios ordinarily used are too small, because they give too little "terminal drop" in the high and intermediate cylinder for best economy.

Fourth, that too little attention is given to the compression curve, which should be determined by the expansion curve, and should never reach initial pressure.

The foregoing is submitted with the full knowledge that the conclusions may not be generally accepted because of a very commendable disposition on the part of the members of our Society to take more kindly to demonstrated facts than to theory, even of the most plausible kind.

While waiting for the verdict of future experiments the author will still further risk his engineering reputation (if he has any) by venturing the prediction, that, when under the conditions assumed on the foregoing pages as to boiler pressure and vacuum, a triple-expansion engine shall be provided with cylinders proportioned to produce approximately the final diagrams of the series in respect to the number of expansions, the terminal drop, and the compression curves, or such slight modification of them as may be suggested by the line of reasoning that has been followed, then a horse-power will be developed from 11 lbs. of steam per hour.

DISCUSSION.

Prof. D. S. Jacobus.—Starting with Mr. Ball's statement that the distribution of steam should be such that each engine works to the best economy between the particular range of pressures selected, which is very nearly so, it remains to determine what should be the distribution in order to obtain the best effect in each engine. Here we do not possess enough experimental data to proceed with certainty. The high back pressure in the high-pressure cylinder of a triple engine brings in an element which makes the case different from that of an engine exhausting against atmospheric pressure, so that it cannot be said that the drop at the end of expansion should be the same as for an engine working under the latter set of conditions. Again, in a triple engine where there is no drop at the end of the expansion lines in the high and inter-

mediate cylinders, the points of cut-off in these cylinders occur at a later point of the stroke than that which has been found by tests to give the best economy for atmospheric back pressure. Mr. Ball, therefore, adheres to one condition found to be economical when an engine exhausts against the atmosphere, but departs from other conditions, so that no definite conclusions should be drawn.

For example, we find in tests made by Dr. Charles E. Emery * that for a pressure of 30 lbs. above the atmosphere the point of cut-off for the maximum efficiency of a non-condensing engine occurs at about .3 of the stroke. This result practically agrees with that found in tests made by Professor Denton and myself, of a 17x30 inch Rand compressor engine used in the construction of the New Croton Aqueduct, in which a few tests were made at 30 lbs. initial pressure. In the Sulzer engine tested by Professor Schroter, the Milwaukee engine tested by Professor Carpenter, and in the Laketon engine tested by Professor Denton, the initial pressure was about 30 lbs. above the atmosphere, and the points of cut-off occurred later than at .3 of the stroke, so that if steam were admitted for a greater portion of the stroke, in order to produce a drop at the end of the expansion lines, the points of cut-off would depart still farther from their positions for best economy with atmospheric back pressure. The same reasoning applies to the points of cut-off in the high pressure cylinders.

It is a fact that in some triple engines the best results are obtained with a load that produces considerable drop in the high and intermediate cylinders. This was the case in a triple expansion engine of over 250 H. P. in use in one of the Edison Company's stations, which was tested by myself at several loads. In this engine the clearance spaces were large, and may have affected the results obtained.

The fact that a particular engine gives the best results when the load is such as to produce a drop in the high and intermediate cylinders, does not, however, prove that such a distribution is desirable, for it may be possible to employ another engine to do the same work with a greater economy.

This discussion is presented not as a criticism on the work done by Mr. Ball, but simply to show that to maintain his position there is a need of further experiments. Mr. Ball, some time ago,

* Transactions, Vol. X., p. 233.

kindly placed one of his engines at my disposal, and constructed special attachments for use in verifying one of his theories, and the outcome is the paper on the effect of compression presented at the present meeting. If we can so arrange, I will gladly undertake tests to demonstrate the truth or falsity of the theory he now presents for some given set of conditions; but the demonstration of the principle, as a whole, can only be accomplished by comparing careful tests of full size engines in actual service.

*Mr. F. H. Ball.**—In considering this paper it must be remembered that the object sought is not to determine absolutely at this time the best cylinders for the conditions that have been assumed, but to illustrate a method by which, with the necessary data in regard to cylinder condensation, the work can be systematically and scientifically done; and that, even if we guess at the cylinder condensation, we can do better than to guess at the cylinders without any particular method, as many of us do. Some general conclusions from the facts that are presented, must be accepted, even by the most sceptical. For instance, with the demonstration before us that a terminal drop of 40 pounds in the high cylinder results in the same proportionate loss by free expansion that accompanies a terminal drop of four pounds in the low cylinder; with this fact in view, no possible condition can be assumed that will permit of no drop in the high cylinder, and a great drop in the low.

As to the best relation between these cylinders in regard to the terminal losses, Professor Jacobus is right in saying that nothing but experiment will settle this question satisfactorily, but I anticipate that the result of experiment will fit the conclusion theoretically reached by the method of investigation under consideration.

I am glad to report that, in addition to the experiments proposed by Professor Jacobus, there is reason to believe that the subject will be still further investigated under the direction of Professor Barr, at Sibley College, and also at the McGill University of Montreal, by Professor Nicholson. Both of these institutions have quadruple expansion engines admirably arranged for experiments of this kind, and the Society may look forward to some interesting data from these sources.

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* Author's closure, under the Rules.

DXC.*

THE THEORY OF THE STEAM-JACKET; CURRENT PRACTICE.

METHOD OF ACTION AND THE SOURCES AND LIMITS OF EFFICIENCY.

BY R. H. THURSTON, ITHACA, N. Y.

(Member of the Society and Past President.)

"J'ai déjà dit que les effets de l'enveloppe, ou chemise à vapeur de Watt, par exemple, avaient été alternativement prouvés ou niés sans que le public parvînt à savoir à quoi s'en tenir sur la réalité des choses."—HIRN.†

INTRODUCTORY.—In an earlier paper the writer presented to the American Society of Mechanical Engineers a summary of facts and current opinion gathered by him from authorities among engineers designing or constructing steam-engines, and among men of science making investigations of the principles and operation of the steam-engine their special task, and collected on both sides the Atlantic by correspondence or study of accessible literature.‡ No attempt was made to summarize the conclusions derived by these distinguished authorities, nor to reconcile apparent differences of experience or of opinion. It was considered wisest to permit the record to stand simply as representative of the opinions of the best authorities of the time, and to allow every reader to draw his own conclusions. It was, however, found that there existed a substantial unanimity of opinion among those best qualified to express their views, whether from the standpoint of experience, as derived by long periods of occupation in designing and constructing, by engineers whose reputation had been built up by years of success in construction, or from that of the man of science, obtaining

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† *Théorie Mécanique de la Chaleur*, 1876, vol. ii., p. 81.

‡ *Transactions American Society Mechanical Engineers*, vol. xii.; Nov., 1890; No. CCCXXXV.

his facts by the study of the principles involved, and by the experimental investigation of the action of the steam-jacket under the various conditions of usual practice.

European engineers and investigators almost unanimously, and the best-known and most successful engineers in the United States generally, though with less unanimity, are convinced that the steam-jacket of James Watt has precisely the place and purpose and fully the relative value assigned it by its famous inventor. It is used very generally in the practice of Great Britain and of the continent of Europe, and much more commonly in the United States, especially in the construction of pumping-engines and other slow-moving engines, than has been usually supposed. The replies to the interrogatories of the writer, as given by the best-known and highest authorities, may be condensed into the statement made by him, at an earlier date, in discussing the theory and operation of the multiple-cylinder engine:*

"*Steam-jacketing* is a common partial remedy for waste. By surrounding the steam-cylinder with the steam-jacket it is possible to produce, in part, the effect of superheating; that is, to secure dryer steam in the engine throughout the stroke. The amount of reëvaporation during the period succeeding cut-off and up to the closure of the exhaust-valve, and the quantity of heat of which the cylinder is thus robbed, measure the amount of initial condensation and waste, and fix the weight of steam which must be supplied in excess of the thermodynamic demand, to compensate that loss. The effect of the addition of a steam-jacket depends upon the conditions of operation of the engine, largely, and may be productive of marked advantage, or, under unfavorable conditions, of no important useful effect. . . . High-speed engines derive less advantage from its application than slow-moving machines; and compound or multi-cylinder engines are less dependent upon it for economy than

* *Transactions American Society Mechanical Engineers*, Nov. 1889; No. CCCXLII., "Philosophy of the Multiple-Cylinder Steam-Engine."

See, also, *Cours de Machines à Vapeur*, Tome II., ch. xxxiv., xxxv.; par M. Haton de la Goupilliere; Paris, 1891.

A most elaborate mathematical discussion of the Hirn experiments at Logelbach, and others, by M. Leloutre, may be found in the *Mémoires de la Société des Ingenieurs Civils*, for September, 1892, and another of almost equal extent, relative to the steam-jacket, in Ledieu's *Nouvelle Théorie Eléméntaire des Machines a Feu*; Paris, 1882; p. 383 *et seq.*

are simple engines. The saving effected in ordinary cases,* by its use, may be taken as averaging about 20%; and about the same gain is attained by effective superheating within the usually practicable range."

* * * * * * *

"It is also seen that any conditions which may make it undesirable, as a matter of finance, to jacket the simple cylinder, would make it still less desirable in the compound or multiple-cylinder engine. As piston speeds are increased, for example, the necessity of the jacket decreases, and the limit at which it will pay to dispense with it is sooner reached in the multiple-cylinder than in the single-cylinder engine. It is this principle which justifies the now not uncommon practice of omitting jackets from marine engines which are driven up to 1,000 feet a minute; while pumping engines, in which the speed is always very low, must always be jacketed, if high duty is demanded."

The data and results of more or less complete and precise trials of engines with and without the steam-jacket, as presented in the first paper above referred to, gave improved performance, as produced by the action of the jacket, ranging from an insensible amount to above 30%; the average of about 100 investigations giving the figure 16%, or, excluding those representing exceptional and impractical cases, in which the engine was operated purposely under conditions far removed from those of daily practice, about 20%. Later experiments with locomotives have shown that it is perfectly possible, as the writer had anticipated, to obtain working conditions under which the jacket would be simply a perfectly useless and undesirable accessory to the engine. The information thus collated showed clearly that the value of the jacket was, as previously stated, greatest at low speeds, and with simple engines of high ratio of expansion. But, curiously enough, the experiments of Donkin and of Kennedy with compound condensing engines, at speeds of rotation ranging from 95 to 100 revolutions a minute, gave some of the highest figures for gain in the whole record, and the later experiments of Professor Alden, at Worcester, similarly show that it is perfectly possible to secure gains of efficiency of very considerable magnitude, and

* "In simple mill engines," is here understood.

in handsomely paying amount, on even high-speed engines, and at approximately 300 revolutions per minute. This consensus of opinion would seem to indicate the conclusion, as derived by experience, that, where the speed of engine is not great, a gain of about 15%, often 20%, may be expected by the use of the jacket on simple engines using saturated or wet steam, and of 10% to 12% on multiple-cylinder engines. In other words, it always pays to jacket such engines — provided the jacket is properly made and operated. Failure to gain sensibly in such cases indicates, not that the jacket is intrinsically useless, but that an inefficient jacket has been applied. A gain much less than 10%, for example, in the common pumping engine, through the action of the jacket, would prove, not that jacketing is a failure, but that that particular *jacket* is defective and a failure.

It is now proposed to study the subject from other points of view.

THE GENERAL PRINCIPLES applying to the case were discovered to be well-recognized and, often, were very definitely enunciated, but by none more concisely than by the inventor of the steam-jacket, James Watt. That great genius discovered the source of enormous wastes in the steam-engine of his time in the condensation of large quantities of steam by contact with the cold internal surfaces of the steam-cylinder at the instant of opening of the steam-valve, and, in less quantity as the piston moved on, up to the point of cut-off. This waste has been since made familiar to modern engineers under the cognomen of "cylinder" or "initial condensation," and still remains the usually most serious of the extra thermodynamic wastes of the machine. Watt's plan for the removal of this source of loss was, as he stated it, "to keep the cylinder as hot as the steam that enters it." He endeavored to reduce the activity of the condensing action by the adoption of the separate condenser, and to check the extent of that action, so far as it remained, by the use of the steam-jacket. He employed non-conducting coverings to protect the cylinder from external loss of heat, and the jacket to check internal wastes, and reduced the tendency to the latter form of loss by, as he said, "suffering neither water nor other substances colder than the steam to enter or touch" the internal cylinder walls. This enunciation of the principles dictating the use of the jacket, often since overlooked by later writers, and in

some instances, ignored, doubted, or even disputed, by authors of recognized authority, stands, to-day, as the first and best expression of the facts and method of jacket-action.

As elsewhere stated by the writer, also: *

"Combes, in papers presented to and published by the *Académie des Sciences*, was probably the first to introduce into the theory of the steam-engine the consideration of that phenomenon discovered by Watt, to check the wasteful effects of which the latter invented the steam-jacket.† That author gradually gave shape to his ideas, as time went on, publishing them in 1845, and, later, in 1863–67. He says in his first paper, just mentioned: 'The utility of the jacket, or, rather, that of heating the cylinders of steam-engines from the outside, . . . is rendered unquestionable, both by direct experiment and by detailed observation of the phenomena characterizing the action of steam in the cylinder, and the logical discussion of these observations.' 'Jackets have not for their main result the maintenance of the temperature of the steam during expansion; their use consists in the prevention of refrigeration of the walls of the cylinder while in communication with the condenser'—probably the first exact statement of this effect ever printed. He even anticipates Rankine and Clausius in one of their most famous discoveries, saying: '*La vapeur d'eau, à l'état de saturation et entièrement sèche, se dilatait sans addition ni soustraction de chaleur; et nous avons montré que l'expansion est alors accompagnée d'une liquéfaction partielle de vapeur. C'est à peu près ainsi que les choses doivent se passer dans les machines à vapeur ordinaires.*' He then goes on to describe very clearly the phenomenon of 'cylinder condensation;' but in his later works he seems to have paid less attention to this action, and may not have fully realized its importance; but his conception of the processes involved in such wastes, and in the preventive action of the jacket, was exact and well expressed. This is Carnot's principle.

"In the once well-known experiments of M. Combes (1843) a Farcot engine was employed under a very light load and with a very high ratio of expansion—about twenty—the trials with and without steam in the jacket occupying three or four working days each. The result was a gain of 36.4% by the use of the

* "Authorities on the Steam-Jacket;" *Transactions American Society of Mechanical Engineers*, No. CCCXXXV., vol. xii., p. 467.

† *Comptes-rendus*, 1843.

jacket, the steam-consumption falling from 40.40 to 30.15 lbs. per horse-power per hour."

According to Mr. Isherwood, "There was no trace of the suggestion until the comparatively recent date of Reech, a distinguished engineer of the French navy, that the surface liquefaction in cylinders was due to the expansion of steam, and was proportional, other things equal, to the measure of that expansion."

M. Dwelshauvers-Dery sums up the work of the earlier continental contributors to the philosophy of the real steam-engine in a series of papers published from 1878 to 1880,* and substantially concludes as follows:

As early as 1843, Combes, taking note of the Watt phenomenon of internal condensation in the cylinder of the steam-engine, proposed the theory of its prevention by the use of Watt's steam-jacket, and indicated that the value of the jacket was mainly to be found in its *prevention* of this condensation during the period of admission. He did not consider it as useful during the period of expansion. Paris, in 1855, reiterates the views of Combes and of Clark, who had, by that time, published his work of earlier date (1850–55).

In 1855, Hirn published a complete study of the action of the jacket, taking a course of investigation which was novel, and previously unthought of. Every phenomenon was carefully observed and exactly weighed, and the laws manifested noted, before attempting their explanation. When he had once proved decisively that the walls of the cylinder were constantly active as "thermal reservoirs," he proceeded to measure their effect, and did so approximately. His method was correct, and he destroyed, utterly, the old idea of the substantial non-conductivity of the interior of the engine. Accumulating evidence and facts, he gradually, from 1855 to 1875, effected the complete conversion of the scientific engineers and physicists to his views and to the truth. The final overthrow of the older theory was completed with the close of the memorable discussion of 1883, in which Hallauer and Hirn and Zeuner took part.

As early as 1851, Lechatelier, Petiet, and Polonceau also indicated the method of action of the cylinder-walls, in the process of waste discovered by Watt, much as did Clark at the same

* *Revue Universelle des Mines;* Liège, 1878–1882.

time; but did not prove their views by experiment, as did the latter. Hirn was the great authority and discoverer in this field on the continent of Europe, and his work remains, and will always remain, a grand monument to his intelligence, his skill, his zeal, and his genius.

Perhaps the first writer to consider and compute the effect of the presence of *water* in the cylinder, in the promotion of initial condensation, and to show that it had precisely the same, and probably a more active, influence on this waste than even the metal of the cylinder-wall, was De Freminville.* His discussion is substantially as follows:

"Suppose the cylinder to contain a certain quantity of water, and let us see what occurs during a single revolution of the crank, neglecting, for the present, the influence of the metallic interior surface of the cylinder, which we will suppose perfectly inert.

"Let n represent the number of cubic inches of water contained in the cylinder at the end of the stroke;

"T = the temperature of that water, which will evidently be that of the entering steam;

"t' = the temperature of the condenser;

"H = the total heat of steam.

"While the cylinder is in communication with the condenser, the mass of water n, in presence of a vapor of low tension and temperature, commences boiling, and a quantity x is evaporated, leaving the water remaining at the temperature t' of the condenser; this remaining mass being $n - x$.

"Then we have the equation

$$nT = (n - x)\, t' + xH;\ x = n\left(\frac{T - t'}{H - t'}\right) \quad . \quad . \quad (1)$$

"There then remains in the cylinder a mass of water $n - x$ at the temperature t', and at that instant steam enters again; coming in contact with the comparatively cold water, it condenses to a certain amount x', making the temperature of the resulting mass $n - x + x'$ equal again to T, and we obtain the following equation of the quantities of heat:

$$(n - x + x')\, T = (n - x)\, t' + x'H;\ x' = (n - x)\,\frac{T - t'}{H - T} \quad . \quad (2)$$

* *Cours de Machines à Vapeur;* Paris, 1862, p. 121.

operating under customary conditions, does not weigh over about 0.01 lb. per square foot, having a thickness of about 0.002 inch as an average throughout the whole area affected. This gives an obstructive power, impeding the passage of heat into and out of the metal, equal to that of a film of iron 0.02 inch in thickness.*

Watt found, as he had anticipated, that the quantity of heat and steam required to simply keep the surface as hot as was the steam about to enter the cylinder, was comparatively unimportant; while the quantity which, without this provision, was condensed, and thus mainly wasted, was very large. The expenditure of a pound of steam in the prevention of the wasting acting was compensated handsomely by the saving of several pounds.

THE CAUSE OF THIS WASTE and of the effectiveness of the jacket being recognized, it is easy to see what are the general principles governing the application of the jacket and the economies to be effected by its use, thus:

(1) The less the intensity of the condensing action in any engine, the less the need and the opportunity for jacketing.

Where, as with very low ratios of expansion, the range of temperature through which the steam expands is small, the resultant chilling action is correspondingly small, and the possible gain by jacketing in proportion. With high speeds of rotation, the time permitted for cooling the cylinder walls is small, and the same results follow. Where the steam enters already amply surcharged with heat, as when "superheated," the initial waste is checked and the jacket has no place or purpose. The compound engine has less to gain by the use of this accessory than the simple engine working under similar conditions of steam-pressure and expansion; and the triple-expansion engine, similarly, can gain less than the compound, other things equal. The "high-speed engine" should be expected to gain less than the engine of moderate or low speed of piston and of rotation.

(2) Intensity of condensing action being the same, that engine which is so designed as to expose smallest areas of surface in its cylinder walls, during the period of its action, will have highest economy, other things equal, and least benefit by the action of the jacket.

* Cotterill, *The Steam Engine*, p. 292.

Thus, long-stroke engines, at any stated ratio of expansion, expose a smaller proportion of cylinder wall to the extreme range of chilling action, and a smaller area producing condensation at the point of cut-off, than short-stroke machines. They therefore lose less by initial condensation and gain less by jacket-action. That engine is least wasteful and has minimum need of jacketing which, at its ordinary and average ratio of expansion and point of cut-off, has the largest ratio of weight of steam present to area of inclosing walls; in other words, the ratio of length of stroke to diameter of cylinder is, approximately, at least, that of stroke to cut-off, if we neglect the surfaces of port passages. As the latter form a decreasing proportion of the total condensing surface, this proportion becomes more exact. Where, as in the best Corliss and other four-valve constructions, these surfaces and "dead spaces" are small, the statement is very nearly true; but in the single-valve engines, with clearances often exceeding 10%, the proportion of stroke to diameter giving least wastes increases.

This is easily shown, algebraically, thus:* The waste-producing area is

$$S = \frac{e \pi d^2}{2} + \pi c d l;$$

$$= \frac{e \pi d^2}{2} + \frac{4 c V}{d};$$

where c, d, l, and V are the cut-off, diameter, stroke of piston, and volume of the cylinder. This becomes a maximum when

$$e \pi d + \frac{4 c V}{d} = 0; \quad d = \frac{c l}{e};$$

e being the ratio of total surfaces of clearance and port "dead-space" to piston and head surfaces alone, a quantity varying from about 1.1 to 2, or even 3, in practice, with, for example, Corliss engine on the one hand, and single three-ported valves on the other. When e may be taken as unity,

$$d = \frac{l}{r}; \quad l = r d.$$

* *Manual*, vol. i., p. 523.

The best proportion, with small clearance, in the common mill engine, with $r = 4$, is a stroke equal to four diameters. In the automatic, with $r = 3$, three diameters. In multiple-cylinder engines, with ratios of $r = 2$ and $r = 2\frac{1}{2}$, the proportions demanded are still further reduced, except as increased clearances tend to produce the opposite result. But the exceedingly long strokes here exacted are impracticable, usually; and one of the special advantages of the multiple-cylinder engine is found in the fact that, with equal clearance fractions, it may be given better proportions for reduction of internal wastes by initial condensation.

The more effective the jacketing or superheating in reducing the wastes by initial condensation, the less is the designer controlled by these considerations in proportioning his cylinders.

(3) The better the quality of the steam, the less the need of jacketing, and the less the gain to be anticipated by the process. With moderately superheated steam initial condensation becomes a comparatively small waste, and the surcharge of heat in the entering fluid performs the office of the jacket, and much more effectively, as a rule. With dry steam the wastes are less than with wet; but it would appear from experiment that the differences in value of various qualities of wet steam may not be important. Superheated steam is a bad conductor and radiator; dry steam, though ever so little above the temperature of saturation, transfers heat with difficulty; but wet steam acts most effectively in the production of this form of loss, taking up and giving up heat with the greatest rapidity. It is of comparatively little use to supply steam only slightly wet; to be of real economical value the process of drying must result in the introduction into the cylinder of really dry steam, and if superheated, though ever so little, the gain may be expected to become very marked. Superheating 100° Fahr. above the temperature of saturation nearly extinguishes initial condensation in most cases. Its effect is more or less satisfactory, as expansion and initial condensation are originally greater or less; but the extent demanded depends obviously on the amount of waste to be checked.

(4) A *secondary effect* of steam jacketing is observed in its evaporation of moisture or entrained water remaining in the working charge after expansion begins. Its efficiency is the

higher as this re-evaporation is effected the more completely, and as it occurs earlier in the stroke.*

Maximum efficiency is attained when all heat supplied enters the system at maximum temperature, and all rejected heat is discharged at the minimum temperature of the cycle. But the heat communicated by jacket to the working fluid is added at less than maximum temperature, and does less than maximum thermo-dynamic work through conversion of heat into the dynamic form of energy. All work performed by conversion of the moisture of the steam into vapor, and by its subsequent expansion, is also effected at less than maximum efficiency. If initial condensation is completely checked by jacket-action, the best possible duty is attained; if, at the commencement of the expansion period, moisture still remains in the working charge, and is evaporated during that period, the earlier this evaporation becomes complete the better the performance of the machine. If, finally, water remains unevaporated at the opening of the exhaust-valve, and draws for its latent heat upon the stored heat in the cylinder walls, all heat thus abstracted is completely wasted.

In the unjacketed cylinder, the exhausted steam contains precisely the amount held in suspension on entering the valve-chest, plus the condensation due wastes of heat by external conduction and radiation, and plus that produced by thermo-dynamic conversion of heat into work.

The jacket must evidently not only keep the surfaces of the cylinder hot and dry, so as to receive the incoming charge of steam without condensing any portion of it; but it must, in fact, also evaporate all of that water of condensation, coming of thermo dynamic conversion with progressive expansion, which may happen to be precipitated upon those surfaces of the cylinder wall. It by no means follows that, as assumed in the Rankine treatment of the jacketed engine, all thermo-dynamic condensation is checked and the steam held in the dry and saturated state throughout the expansion period. Much of this water of condensation is undoubtedly suspended in the mass of the working steam as mist, and escapes to the condenser as water; but portions in contact with the cylinder wall must be re-evaporated if the surfaces are to be held up to the tempera-

* Cotterill considers this the real use of the jacket.

ture of prime steam, and kept dry. This must be disposed of by jacket-heat before the steam-valve again opens; as must all water lying in pools, in cavities in the cylinder port spaces, or clearances. Any water anywhere present in the working cylinder, at entrance of the steam, must absorb so much heat as is needed to bring its temperature up from that of the condenser, approximately, to that of boiler-steam, nearly. Having eight times the capacity for heat possessed by iron, the presence of a very small quantity, comparatively, may do great harm. With large total ratios of expansion, the water of thermodynamic condensation may prove a sensible burden upon the jacket, and apparently slight defects in design may permit still heavier taxation by causing settlement of water in the cylinder spaces.

The action of the jacket being one of preparation of the cylinder walls, during the expansion and exhaust periods, for the reception of the succeeding charge of working steam, it is seen that the jacket may be effective, even though not transferring heat through the cylinder wall during the period of introduction.* In other words, it may be advantageous, but certainly is not essential, that the pressure and temperature of the steam in the jacket should be higher than in the cylinder before the close of the cut-off valve. The one essential need is that the heat transferred from the jacket shall be sufficient to insure dryness and maximum temperature of the cylinder wall in advance of the entrance of the charge. Transferring less heat, the jacket is inefficient; transferring more, it is needlessly wasteful. If higher pressure will, in any given case, insure the proper heating of the surfaces, without an excess productive of unnecessary waste during the exhaust period, it will prove advantageous; otherwise not. In fact, could the jacket be made or applied in such manner that its action could be suspended from the instant of closure of the cut-off valve to the end of the exhaust period, and then made instantly to bring up the temperature of the cylinder wall to that of boiler-steam, its function would be fulfilled with ideal perfection. As it is, its work must be performed during the periods of expansion and exhaust.

It is clear that each pound of steam condensed in the cylinder

* As the writer has sometimes stated it, the task of the jacket is simply that of maintaining the temperature of the cylinder wall.

is twice evaporated; once in the boiler, once in the engine; but each pound condensed in the jacket is but once evaporated—in the boiler.

The condensation by action of the cylinder walls does not affect the quality of steam finally discharged into the condenser. All steam thus condensed is re-evaporated by abstraction from the surfaces of the cylinder of all the heat stored in them during the period of initial condensation.

In the jacketed cylinder, the water converted into steam in the cycle may exceed the quantity condensed at entrance if the charge be then wet, and the jacket may thus produce, during the later part of the expansion and the whole of the exhaust period, waste otherwise not liable to take place. This waste will be measured by the excess of water re-evaporated over weight of steam initially condensed. It may be readily imagined that there is a possibility of the jacket proving, under some such special conditions, a wasteful, rather than an economical, adjunct to the steam-engine. Its final efficiency and that of the engine will depend largely upon the completeness and the promptness with which its work is performed in holding up the temperature of the surfaces in contact with steam at cut-off to the temperature of prime steam, and in drying them off, if wet at the beginning of expansion.

(5) The value of the steam-jacket depends upon its location or distribution over the cylinder walls, and upon the ease with which heat can pass from jacket to cylinder during the period of introduction of the charge.

Jacketing the heads, the piston, and the surfaces of the cylinder up to the point of designed cut-off, obviously may be expected to give best disposition of jacket and best action. Jacketing the body of cylinder between the two points of cut-off of opposite ends, is as obviously comparatively useless. By jacketing is, of course, meant *effective* jacketing. Piston jackets have rarely proved effective, and the jackets on many engines are inefficient throughout; probably the failure to distinguish between the uselessness of all useless jackets, and the uselessness of some well-made jackets, may have had much to do with the prevalent prejudice against their proper employment. Many jackets fail to do effective work through defects of design, construction, or, more usually, of operation, and these cases have done much to discourage the use of that appurtenance. If

jacketing is to be adopted at all, it is evident that the heads and port-spaces are the most essential areas to be covered by the jacket. Pistons are so difficult to jacket, and, especially, to thoroughly drain their jackets, that few designers would attempt to thus cover them; although the importance of their surfaces as condensing areas, and the need of jacketing them, may be seen to be quite as great as with the cylinder heads. The greater the diameter of piston in proportion to stroke, the greater the importance of jacketing heads as compared with covering the sides of the cylinder with the jacket.

A correctly located jacket may be ineffective in consequence of the difficulty of passing heat from it into the cylinder promptly and in ample amount. Thick walls, the presence of core-sand or other non-conducting obstruction, defective drainage, the presence of air in the jacket, all these are ways in which the best arranged jackets, in all other respects, may be made comparatively, if not absolutely, useless; and such defects have done much to bring steam-jackets into discredit where, otherwise, they might have greatly improved the economic performance of the engine. Instead of seeking the reason of the apparent failure of the jacket, it has often happened that the observer condemns jacketing, off-hand, rather than the defective jacket.

It is now well ascertained that smoothly finishing the cylinder heads and the faces of the piston, and even polishing them, may prove handsomely remunerative by checking these internal thermal wastes. This probably occurs both by reducing the heat-absorbing power of the otherwise rough surfaces of the castings, and by making it easier for the jacket to hold up their temperature to a maximum. A rough-cast surface may have much greater active area than if smooth; every asperity adding surface area.*

As the writer has remarked elsewhere:† "A well-polished internal surface, especially if bathed in oil, is hardly less effective in reducing wastes than is well-dried steam."

(6) The gain by a good jacket depends upon the quantity of steam acted upon and the time of action.

The less the quantity of steam in the charge, other things

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* *Transactions Inst. Naval Architects*, 1894. D. Croll.

† *Manual of the Steam Engine*, New York, 1891; Vol. I., p. 537.

equal, the more effective the jacket in drying it, and keeping it and the cylinder walls in contact with it dry. In other words, the greater the ratio of expansion the more useful the jacket—a well-known fact of experience. The more time allowed for its action, the more complete is that action. In other words, low-speed engines derive more benefit from the jacket than high-speed; which is another familiar fact in every-day experience.

The time of action, it should be noted, is not that of entrance of the steam into the cylinder; it is that allowed for the preliminary drying of the internal surfaces of the cylinder and warming them up as nearly as possible to the temperature of the prime steam, in order that, when steam does enter, it shall find those surfaces hot and not condense upon them. As in the production of internal wastes, their amelioration is a function, not of time measured from the beginning of stroke to point of cut-off, but of time elapsing from cut-off to the beginning of the following forward stroke. The most serious refrigeration, during this period, probably occurs during the exhaust stroke. The work of the jacket is that of prevention, not of remedy; of preparation, not of restoration of the conditions favoring economy.

(7) A good jacket involves provision for free introduction of steam into the jacket-space, perfectly free and thoroughly complete drainage of the condensed water back into the boiler, and the most rapid possible flow of heat from jacket to working steam.

Defective steam-supply, and, oftener, ineffective drainage, have probably been the most usual causes of the inefficiency of otherwise well-located and desirable jackets. Entrapped air has sometimes been responsible for the condemnation of a jacket. Wasting steam from jacket traps or loss of the water of condensation—a very considerable item, if the jacket is doing good work—sometimes accounts for an undeserved reputation for uneconomic action. Free supply of heat from jacket to steam is effected by using a thin intermediate wall of good conducting material—wrought iron and, more usually, steel being generally used for this reason in preference to cast iron—and by securing an ample temperature-head between the two masses of steam. With wet steam, the waste during exhaust may exaggerate loss more than the gain by the jacket during the remainder of the cycle reduces it, in some cases; in which event the above state-

ment becomes subject to reservation. But the use of wet steam is inexcusable, in most instances, and the jacket cannot be condemned on that account.

The general principles above enunciated are sufficiently obvious, once the purpose and method of action of the steam-jacket are fully recognized; but the limitations of its action still remain somewhat obscure, and it is still uncertain, in many cases, what construction of jacket, if any, should be adopted, and how much advantage may be fairly expected to result from its operation. General experience has thrown much light upon the subject, and we know well, in most cases, what to anticipate; but while we find that it may, with simple engines, with high ratios of expansion, give large gains, we also know that it gives comparatively little advantage in multiple-cylinder engines, and that, especially when the speed of the engine is high, in both cases, the gain by its use is comparatively small. But it still remains to be ascertained where, in doubtful cases, the line shall be drawn between the paying and non-paying use of the jacket, the point at which the costs of its adoption are such as to cause it to fail to pay fair returns on that small investment. It can be seen, however, that a very small percentage of increased economy, only, is required to make it pay well.

THE ACTION OF THE JACKET, in detail,* is probably not complicated; but it is obscure because of the facts that it is so far out of reach of the investigator that the variations of temperature and of heat storage and transfer affect variable quantities of metal and fluid which the engineer cannot easily measure, and are subject to intricate and uncertain physical changes of condition and quality of the mixture of steam and water, or possibly, at times, of dry and superheated steam, similarly difficult of determination.

(1) *Jacket and cylinder receive gaseous steam; i.e.*, the fluid is highly superheated and behaves like a gas.

In this case the action of the jacket tends to keep the inner walls of the cylinder up to its own temperature. Assume this possible. The gaseous steam enters the cylinder at maximum temperature, expands, doing work, constantly losing both heat and temperature, down to a minimum, at exhaust, and is finally discharged, it may be assumed, dry but saturated. Each enter-

* From *Manual of the Steam-Engine*, § 153, p. 627.

as heat; and the proportion of these two quantities—the partial waste by inefficient transformation and the pure waste—is determined both by the extent to which expansion is carried and by the quality of the working fluid.

If the steam be dry, or nearly so, at the close of the first period, and if the second, the expansion period, is sufficiently prolonged, the action of the jacket and the heat-storing property of the metal of the cylinder promptly results in superheating the expanding steam and so checking further waste of heat from jacket and from cylinder walls during the terminal period of expansion and during the exhaust, and thus allows the jacket to raise the temperature of the cylinder promptly and fully to that of the entering steam. This being accomplished, initial condensation is, in turn, reduced to an unimportant quantity; the total waste is mainly jacket-waste, and is a minimum.

On the other hand, if the amount of water produced, either by initial condensation or by the work of expansion, or both, is so great that it cannot be all re-evaporated early in the stroke, and if the cooling of the cylinder walls is thus continued, the jacket-waste becomes increased, the waste which it is intended to check may remain serious, and the result may be a considerable net loss, and but little or no advantage from the jacket.

This must be the result, probably, to a greater or less extent, whenever the drying of the cylinder and steam is not nearly or quite completed at the opening of the exhaust-valve, as when the jacket is defective or the steam too wet. It would seem possible that intermediate conditions might prove to be those of best jacket-action.

The process is here, probably, one in which the first effect of the jacket, during expansion, is to dry the steam, which contains always, if not superheated, suspended within its mass more or less of the water of initial condensation; next, the checking of condensation due to the work of expansion; and, finally, the superheating of the steam, if the earlier stages are completed early enough, and existing conditions permit. The first portion of this process gives a gain of work by adding steam to that existing, as such, at the beginning of expansion; the latter portion by giving the steam larger work-power.

During the exhaust period there is a pure waste of heat, with a compensating gain by drying and heating the interior surfaces of the cylinder preparatory to the entrance of the next charge

of steam. Compression has a similar effect, as a result of the conversion of the work of compression into heat.

(3) *Wet steam is supplied.* In this case, the jacket, on its side, acts precisely as before. The water in the steam in the jacket drains out or is trapped off, and is returned to the boiler, leaving the steam practically dry, as before. But the interior of the engine is placed under quite different conditions.

In addition to the heat demanded of the jacket to keep the working steam dry, and to first dry off and then warm up the interior surfaces of the cylinder, a quantity of heat which, within limits, will be larger as the steam is initially wetter, and which may be often very great, is drawn from the metal and from the jacket, throughout substantially the whole cycle, to evaporate all or a part of the entrained water, and to *then*, if possible, dry off the metal and to heat it up again to the maximum temperature. Not only is this amount of heat increased with increase in quantity of water entering with the steam, but the proportion of heat drained off wastefully in the terminal portion of the expansion, and throughout the exhaust period, is increased as the quantity of water to be evaporated is greater; so that it may readily be believed that the interior of the cylinder, drenched and flooded with water at the opening of the steam-valve, may continue to act as a waste-producing boiler quite through the cycle, thus causing great loss during the exhaust period; when, the difference of temperature being a maximum, the heat which the jacket is capable of thus wasting becomes itself a maximum, and both absolutely and relatively large.

If the water of initial condensation is not, in any instance, all re-evaporated during the expansion period, it will be re-converted into steam during the exhaust period.

It is thus obvious that the quality of the boiler-steam is important; and it may be seen that dry steam is an element of successful action of the jacket. It is as unquestionably the fact that dry steam is essential to the best action of the jacket, as that superheated steam may render the jacket unnecessary and useless.

General Experience, at the moment, is our only reliable source of information, and the designer must content himself with the best study that the records permit, as a means of determining whether he should, in any proposed design, make use of

this accessory. With slow-speed engines, as most steam pumping engines, he will never hesitate, probably; with mill engines, also, of moderate speed, whether in either type, simple or multiple expansion, he will rarely question the desirability of jacketing, as seen from the standpoint of finance. In the case of high-speed engines of any class, simple or multiple, he may often be unable to decide where the value of this expedient for increasing efficiency becomes insensible, and, doubting, will usually, probably, decline to use it, on the ground of first cost, complication, and possible increased expense for maintenance.

Watt supplies the first essential fact, in a statement already quoted in the preceding paper by the writer on this subject. The quotation will bear repetition.

He says, in a letter to Dr. Small, as early as in 1769:

"I have misimproved the cylinder by making the bottom of cast iron, which . . . does not communicate heat fast enough to evaporate the water left in the cylinder the first stroke, without being considerably cooled by it, and consequently condensing as much the next. I have also done wrong in removing the outer cylinder, which kept the inner one always surrounded by steam, ready to supply any loss of heat that might happen by evaporation from the inside surface."

He proceeded to correct these mistakes, and later, writing to Dr. Roebuck (1769), he explains his ideas more fully. He says:

"At first, when the cylinder is cold, there is a small quantity of water left in the cylinder. Immediately on producing the vacuum the water is converted into steam, and thereby cools the cylinder considerably. Now, if there is an external cylinder, and the bottom of the internal one is thin, the steam on the outside will warm it again instantly, and no steam will be condensed the next time. (This was the case in my last model.)"

In giving an account of his experience, he says: "When we tried to lay aside the jacket we had no reason to applaud our economy, for the consumption of fuel was considerably greater."*

* Robert Stuart says: "It is admitted that Watt had an engine at Soho, acting on the expansive principle, as early as 1776. In 1778 he erected another on the same construction, to pump water for a public company in London; and although another mechanic, three years after this later machine was in operation, took out a patent for a similar contrivance, we will describe it as an invention

The testimony, supplied in ample amount in the same paper, shows beyond shadow of doubt, and the fact is now never disputed by engineers familiar with the subject, that the steam-jacket, in the engines standard until recently, with moderate piston speeds and with ratios of expansion ranging from six upward, whether simple or compound, produced a decided benefit, in the best examples amounting to 15% to 20% in simple engines, and to 10% to 12% in compounds. In European practice, the use of the steam-jacket on such engines, where economy of steam and fuel is sought by good designers and constructors, is almost universal. In the United States, where condensation is less common, and lower expansions prevail, with, usually, higher piston speeds, the facts of the case are less familiar as matters of experience. In special cases, gains of 25, 30, and even 35% or more are recorded, where the ratios of expansion are exceptionally great, and the working conditions unusually favorable to the introduction of jacket-action.

A good jacket-action is even found to be obtainable at high speeds. Thus, one case is reported in the *Journal of the Franklin Institute* of 1859, quoted by the committee of the British Institution of Mechanical Engineers, in which a gain of 23% was attained at a speed of 203 revolutions a minute.* Brodie's experiments give an average gain of 13.5%, at speeds ranging from 92 to 102 revolutions a minute.* Kennedy (1889) reports an average of 22%, at speeds of 96 to 101 revolutions.* Brodie's point of cut-off was about one-third; Kennedy's averaged about one-eighth. The latter attained above 35% at 9 and 11 expansions. But the most remarkable instance of successful jacketing at high speed is found in the engine of Professor Alden, at Worcester, in which, at 290 revolutions per minute, a gain of about 15% is obtained by the use of this simple device; the jacket demanding about 7% of the steam supplied. The steam used in the jacket is thus twice as efficient as that employed in the cylinder. Mr. Bird's analysis of the action of the jacket for this case, by the Hirn system of investigation, shows that the cycle is made more

worthy of Watt's genius, and which only his wonderful skill could have called so perfectly into existence." This invention was described in the famous patent of 1769. The above facts have sometimes been denied, but even the Newcomen engine, in working properly, must cut off steam before the end of the stroke. The higher speeds and pressures of Watt's engine rendered this still more imperative.

* "Authorities on the Steam-Jackets," Appendix; *Transactions A. S. M. E.*, Vol. XII., p. 500.

nearly adiabatic in its expansion period, the initial condensation is greatly reduced, the drying of the cylinder by re-evaporation commencing earlier, and the exhaust waste is decreased.* These results are the more instructive for the fact that that engine was of but 10 H. P., and compound.

It will be seen that the efficiency of the jacket depends upon the quantity of heat it can transfer and steam it can condense per unit of area of its useful surface. Restriction of steam consumption in the jacket is only desirable by restriction of the area of wasting surfaces which it should cover, as by reducing the areas of cylinder walls in the clearances and port-spaces. Small gain by the use of the jacket, in cases where the consumption of steam by the jacket is also small, may always be taken as *prima facie* evidence of defective jacketing, or of the use of the jacket where other special methods of effecting economy have made the jacket superfluous. This is particularly the fact where the expenditure of heat and of steam in the jacket is less than the reduction of wastes, or where, as sometimes is the fact, the consumption of steam in the jackets accounts for but little more than the waste by external losses of heat. The case of the often-quoted Pawtucket pumping engine, in which an expenditure of 5% in the jackets only gave a net gain of 3%, under conditions almost invariably resulting in very marked economy by its use, is no evidence of the uselessness of jackets, but simply proves that jacket to be, for some reason to be yet ascertained, comparatively useless. It evidently supplies little more than the heat demanded for external wastes, and gives but very little to the working steam. The correct deduction from that case is that the reduction of external wastes, of heat to zero, in their action on the working charge may cause sensible economy, by ameliorating the tendency to initial condensation, coming of that method of cooling of the working cylinder. It shows the importance of carefully covering the exterior of the engine with good non-conductors, to avoid that exaggeration of the wastes by initial condensation which is consequent upon external cooling. The equally obvious deduction from general experience in the use of jackets is that, with the exception of the so-called high-speed engines and those multiple-cylinder engines in which all recognized elements of highest

* "Effect of Steam-Jacket on Cylinder Condensation;" *Trans. Am. Soc. M. E.*, Vol. XII., 1891, No. CCCCLIII.

efficiency make the jacket superfluous, its use may be expected to effect gains which will always more than compensate their costs, and usually may yield large returns upon the expenditures incurred, directly and indirectly, in the introduction of that element of the machine. Failure to secure such returns proves, as has just been stated, not the inefficiency of jacketing, but the inefficiency of the particular jacket thus failing to do the work assigned it.

The indications from general experience would seem to be that the jacket, when working effectively, ought to exhibit maximum economic result when it reduces the internal wastes of the engine to about one-half their amount in the same engine, unjacketed. In so far as the jacket fails of this activity it may be pronounced defective; more than this seems more than can be expected of it, notwithstanding the fact that cases have been sometimes reported in which it has been stated that initial condensation has been entirely prevented. Mr. Isherwood thinks that it can be shown that this limit of maximum action can be deduced by rational investigation; but of this there may be some doubt.* In his view, however, this "percentage is at once the measure and the limit of its economic effect."

Experience shows the jacket to have some advantages apart from its thermal action upon the cycle of the engine, and incidental to its construction and operation. It permits the gradual heating up of the engine, preliminarily to starting, and thus enables the engine driver to evade those risks which come of sudden and irregular heating of large masses of cast iron, and those due to the formation of masses of water in the cylinder at the entrance of hot steam into a cold metallic chamber, and which have been known to produce bad accidents when not very skillfully dealt with.

In the Worthington type of direct-acting pumping engine, this fact gives a special advantage when the jacket is used, in permitting the completion of the stroke at higher pressure than otherwise, and giving steady motion of engine. The uniform temperature insured by the jacket keeps the cylinder in correct form, and prevents that waste of power and wear of cylinder sometimes attributable to the well-established fact that its bore

* *The Steam-Jacket, its Genesis, its Purpose*, etc.: B. F. Isherwood, Chief Engineer U. S. N., Engineering Cong., Chicago, 1893; pp. 40–43.

may be, when in operation unjacketed, smaller at the middle than at either end; the mean temperature being necessarily higher at the ends than at the middle, and this difference becoming very sensible with large ratios of expansion in a single cylinder.

Experience, on the other hand, also shows that the jacket may have serious disadvantages incidental to construction and use; although they commonly come of defective original design or construction, and are not, as a rule, necessarily inherent in the system of application of steam-jackets. If not properly connected with the body of the cylinder, or if made by the introduction of a "liner" into the cylinder, leaks may occur of sensible, or even of serious, amount, and the value of the jacket may be thus made a large negative quantity. Core-sand left in the spaces which should be filled with steam, or covering the surfaces of the cylinder walls, where they should be of the highest possible heat-conducting power; entrapped air, taking the place of the steam; ineffective drainage, filling those spaces with water; shrinkage stresses, liable at any time to produce fracture by the irregular changes of form of the complicated mass; any one of many possible defects of design, construction, and manipulation may be responsible for disappointments in the use of the jacket, and for frequent unsatisfactory reports in regard to its economic effect. The final result of experience to date may be summed up in the statement that when a jacket is properly designed, constructed, and attached, and when it is properly manipulated, it may always be expected to give good economic results in simple engines, and large returns on the investment, with all classes except possibly the high-speed engines. In exceptional cases, however, good results have been obtained and handsome returns, with speeds as high as 200 revolutions and upward per minute. It still remains a question whether the unsatisfactory reports from its action on fast-running engines may not be attributable to defects in jacketing rather than to limitation of the system to slow engines. The same remarks apply to the case of multiple-cylinder engines, which have often exhibited excellent results, although both the high-speed and the compounded engines are obviously less promising of gain than other types of engine. Where superheated steam is employed it is probable that no advantage will be found economically in the application of the jacket.

Wherever applied, it is evident that it is very desirable that the engine should be supplied with dry steam in order that the work of the jacket may be made a minimum; that heads as well as, or in preference to, sides should be covered; that clearances and port-passages should be made the smallest possible; and that it should be at all times possible to see that the jacket is in action, and in active operation as well. Correct design, construction, and operation will usually insure a very large and satisfactory return for all costs and time and thought expended in insuring its effective use.

THE THERMO-DYNAMIC THEORY OF THE JACKETED ENGINE, as customarily constructed, following its first writer, Rankine, involves assumptions which are not found to correspond with general experience and usual practice; but which, at least, give definite form and exact solution to the problem as thus treated. The primary assumption is that the jacket yields heat to the working charge of steam in just sufficient amount to hold initially dry steam in the dry and saturated state throughout the working stroke, leaving the surfaces dry at its end, and heated at the end of the cycle to the temperature of prime steam. This assumption is supposed to be justified by the well-known fact that, while wet steam takes up heat greedily, and thus promptly becomes dry and saturated when supplied with the needed heat, dry steam absorbs heat but slowly, and superheated steam is a non-conductor and non-absorbent of heat. Hence it is possible that the cylinder walls, if held up to the temperature of boiler-steam by the jacket, would dry the entering steam promptly, if approximately saturated, and would communicate heat to the expanding charge with rapidity when wet, and slowly or even imperceptibly when dry; thus, if the cylinder walls could be held up to maximum temperature, insuring the continuous retention of the working steam in the dry and saturated state, in spite of the tendency to become wet through the action of thermo-dynamic conversion and partial condensation, actually observable in considerable amount in the non-conducting or the unjacketed conducting cylinder. Were this assumption to be accurately true, we should have an exact theory of the jacketed engine.

Experience and direct experiment unite in showing this assumption to be inexact, however; and in spite of the action of the jacket, in all usual constructions of engine, the steam becomes

wet at the start by initial condensation and remains more or less wet in most, though not in all, cases, up to the end of the expansion period, and even, in many instances, into or through the period of eduction. The theory of the ideal jacketed engine, like that of the ideal unjacketed engine with its non-conducting cylinder, requires to be supplemented by what Hirn called the "experimental theory," by a theory of thermal wastes, internally and externally, and of dynamic wastes, to render it available for the purposes of the engineer. The two representative ideal cases will be discussed in summary; and this discussion will be followed by the examination of the real case.

THE IDEAL STEAM-ENGINE, to be studied in comparison with the real engine, is, properly, the case of the non-conducting cylinder, in which no thermal wastes can occur, and which, therefore, requires no such artificial adjunct as a steam-jacket, and only employs superheated steam as giving, by its higher temperature range, a larger conversion of heat into work than the engine using saturated steam. This case has been often studied, and is familiar to every student in engineering and every thermodynamist. Adopting Rankine's methods of treatment, we obtain the following results in computing its efficiency, in a set of examples fairly representative of current practice: *

EFFICIENCY OF STEAM—IDEAL CASE.

Non-conducting Cylinder.

p_1 = absolute pressure of admission;
p_2 = absolute pressure at end of expansion;
p_3 = mean back absolute pressure;
τ_1 = absolute temperature at p_1;
τ_2 = absolute temperature at p_2;
τ_4 = absolute temperature of feed water.

Then the energy per cubic foot of steam admitted:

$$U D_1 = J D_1 \left\{ \tau_1 - \tau_2 \left(1 + hyp\ log\ \frac{1}{\tau_2} \right) \right\} + \frac{1 - \tau_2}{\tau_1} L_1;$$

$$+ r (p_2 - p_3);$$

* "Maximum Steam-Jacket Efficiency." R. H. Thurston. *Journal Franklin Institute*, April, 1891.

where

r = ratio of expansion;
H_1 = total heat of evaporation;
D_1 = density of steam at p_1;
L_1 = latent heat of steam at p_1.

Heat expended per cubic foot of steam admitted:

$$H_1 D_1 = J D_1 (\tau_1 - \tau_4) + L_1;$$

$$Eff.\ of\ steam = \frac{U}{H_1}.$$

Assume an initial boiler and engine pressure of 115 lbs. per square inch, absolute, and a back pressure, in cylinder and condenser, of 4 lbs.; with varying ratios of expansion, we obtain the following values of the thermo-dynamic efficiency of the ideal engine, constituting the limits which we seek to approximate in the real case, but which we can never hope to pass: *

EFFICIENCIES OF WORKING FLUID.

IDEAL STEAM-ENGINE, UNJACKETED.

Cut-off.	Ratio of Expansion.	Efficiency.	Steam per H. P. Per hour.	B. T. U. PER H. P.	
				Per minute.	Per hour.
0.05	20.00	0.2073	11	202	12120
.10	10 00	.1934	12	222	13342
.15	6.66	.1795	13	235	14139
.25	4.00	.1566	14	270	16210
.35	2.85	.1358	16	310	18712
.45	2.22	.1237	19	342	20524
.55	1.82	.1119	21	379	22723
.75	1.33	.0898	26	471	28278
1.00	1.00	.0707	33	597	35845

These expenditures would be, ordinarily, doubled in the higher ranges of expansion, in even good practice, by extra thermo-dynamic wastes, in simple engines, and increased 50% in compound and one-third in triple-expansion engines, assuming all to be unjacketed.

THE IDEAL CASE, WITH JACKETED ENGINES, gives the following efficiencies for the same condition of initial and back pressure. The symbols and formulas are:

* *Manual of the Steam-Engine.* R. H. Thurston. Vol. I., § 155, pp. 640, 641.

V_1 = volume of 1 lb. steam at p_1;
V_2 = volume of 1 lb. steam at p_2;
a = 1109550 foot-pounds.
b = 540.4 foot-pounds per degree Fahrenheit.

Then the energy exerted is:

$$U = a\, hyp\, log \frac{\tau_1}{\tau_2} - b\,(\tau_1 - \tau_2) + V_2(p_2 - p_3).$$

Whole expenditure of heat per pound steam:

$$h = J(\tau_2 - \tau_4) + a\left(1 + hyp\, log \frac{\tau_1}{\tau_2}\right) - b\,\tau_1, \text{ nearly.}$$

Efficiency:

$$Eff. = \frac{U^1}{h}.$$

EFFICIENCIES OF WORKING FLUID.*

IDEAL STEAM-ENGINE, JACKETED.

Cut-off.	Ratio of Expansion.	Efficiency.	Steam per H. P. Per hour.	B. T. U. PER H. P.	
				Per minute.	Per hour.
0.05	20.00	0.1930	12	213	13167
.10	10.00	.1808	13	234	14085
.15	6.66	.1665	14	254	15239
.25	4.00	.1442	16	295	17674
.35	2.85	.1302	18	326	19578
.45	2.22	.1209	19	355	21283
.55	1.82	.1087	21	388	23312
.75	1.33	.0812	27	524	31420
1.00	1.00	.0707	33	602	36126

These expenditures of heat and of steam would be increased by wastes, in the real engine of which this is the ideal representative, by from 50 or 60% at the higher ratios of expansion, and for best practice, to 20 or 25% at low ratios, if simple engines are compared, and by about one-half those figures for the compound engine or one-third for the triple-expansion. Comparing these ideal cases, however, it is at once seen, as could easily have been predicted, that the jacketed engine is the more inefficient of the two, and by about 1% loss of efficiency through-

* *Ibidem.*

out the whole usual working range of modern engines. In the cases referred to:

"An examination of the tables, of the curves still better, will show clearly the wasteful influence of the steam-jacket, as an element considered by itself. Within the usual range of practice, from about 5 or 6 to 15 or 20 expansions, under the assumed conditions of initial pressure and cut-off, it is seen that the loss by its application is fairly constant at something above 1%, in these cases, rapidly falling to zero as the ratio of expansion falls from the lower figures to unity. The consumption of steam in pounds per horse-power per hour may be computed very approximately by dividing 2.5 by the computed efficiencies. The cases assumed are for condensing engines, and the evaporation always taken at 9 lbs. of steam per pound of fuel, the fuel expenditure may be gauged by dividing the weight of steam computed by nine. This gives, for example, about 12.06 and 12.95 lbs. for the unjacketed and for the jacketed engine, respectively, at a ratio of 20, in steam demanded; and of about 1.33 and 1.44 lbs. of fuel. For a ratio of expansion of 4, the figures become about 16 and 17.3 respectively for the steam, and 1.75 and 1.85 lbs. of fuel. At full stroke, the figures become 35 lbs. of steam and of feed-water, and 4 lbs. of fuel per horse-power and per hour, for both engines.

"We conclude from the above, therefore, that, in the ideal case, the steam-jacket reduces efficiency, necessarily and without exception, and that, for ordinary variations of the ratio of expansion for the pressure here assumed, this waste is a nearly constant quantity. To put it more mathematically, the gain is negative and nearly constant at about 1% loss of efficiency." *

"This is sufficiently evident, *a priori*, from the consideration that the latter receives all its steam at a maximum temperature, expands it adiabatically to a certain terminal temperature, and then exhausts it; while the former receives a part of its heat at intermediate temperatures, expands the fluid non-adiabatically, and finally rejects it at the terminal temperature, with a lower mean range of expansion. In other words, the jacketed engine departs furthest from the principles of economical operation first enunciated by Carnot: † All heat should be received at

* *Ibidem.*

† *Reflections on the Motive-Power of Heat.* By Sadi Carnot. Edited by R. H. Thurston. New York: J. Wiley & Sons. 1890.

maximum temperature; expansion should be perfectly adiabatic, and should continue to the minimum temperature and pressure, and all should be rejected as nearly as possible at that minimum.

"The fact is thus sufficiently well shown that the steam-jacket, as employed on the steam-engine, of whatever form and arrangement, is intrinsically a wasteful element, and that its use only gives, in certain cases, an economical advantage by its repression of wastes of larger magnitude. It checks a serious and unavoidable waste more or less completely, by a process which as inevitably involves a waste which is commonly, but, perhaps, not invariably, a lesser one. The ideal steam-engine, such as is treated of in the purely thermo-dynamic study of the steam-engine, has a lower efficiency with, than it has without, a jacket."

The intrinsic waste of the steam-jacket may thus be taken as,

EFFICIENCIES OF JACKETED AND UNJACKETED ENGINE.

Steam, 115; back-pressure, 4; ratio of expansion variable.

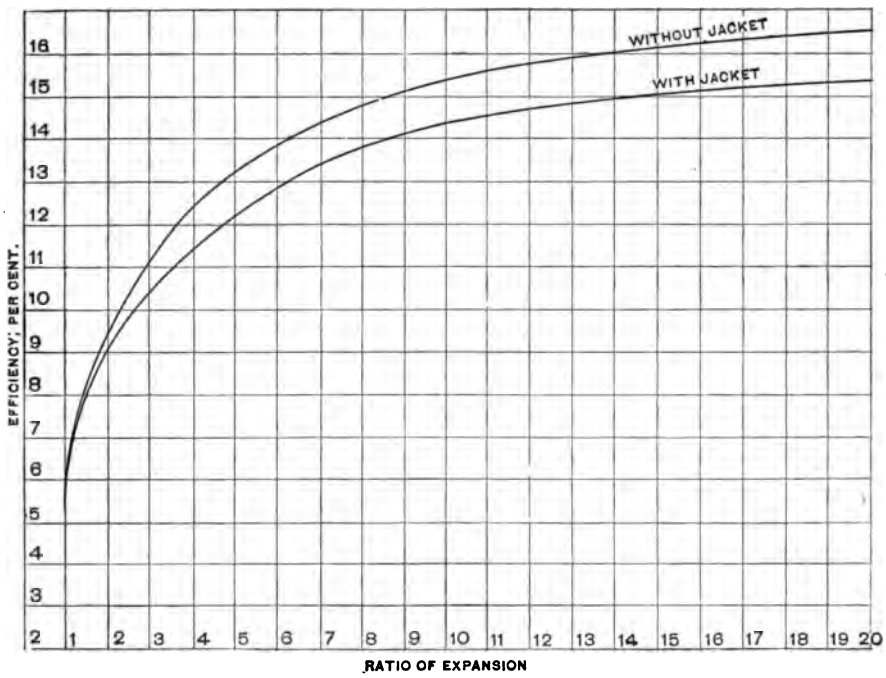

FIG. 243.—Effect of Jacket.

in usual good practice, about 1% of the thermal equivalent of the work done for efficiency unity, or from 6 to 10% of the costs of power, in steam and fuel, in the real engine of common good practice. Its use, to be of service in promoting economy, must reduce the wastes of the real engine by more than these quantities; and its actual economic effect in diminishing internal wastes are measured by adding to the reported gains, as shown by engine-trials, these figures for external waste. That is to say: where the apparent gain by the addition of the steam-jacket is 20%, the account is actually made up of a credit of 25% or more, partly canceled by a debit side of 5% or more. If it cannot reduce or prevent extra-thermo-dynamic wastes by more than this 5% the net result will be a loss, in such case.

The jacket is also chargeable, in the real engine, with exaggerating the losses of heat due to external conduction and radiation; amounting usually to 1% and upward, often to over 5%, of all heat expended; all of which heat is abstracted from the jacket, when applied. Its intrinsic waste of 1% efficiency may be reckoned as 0.025 lb. of steam, nearly 25 B.T.U. per horse-power and per hour, for efficiency unity, or from one-sixth to one-fourth of a pound of steam, and from 175 to 250 B.T.U. for engines ranging between 10 and 15% efficiency; in which 6 to 10% of the heat supplied is thus used. External heat-wastes vary nearly inversely as the linear dimensions of the cylinder; but, in engines of moderate size, usually consume one-sixth to one-tenth as much as the jacket action. In other words, one-sixth to one-tenth the minimum heat supply to the jacket is required for external, five-sixths to nine-tenths for internal, waste; the sum amounting, usually, to somewhere about 10% as a minimum in the simple engine; the cycle being such as is assumed in the Rankine theory of jacket action. These wastes are reduced by external coverings of non-conducting materials and by every circumstance which favors the restriction of internal jacket-wastes. The theory giving this computed 1% difference of efficiency against the jacket assumes that it extinguishes all other internal extra thermo-dynamic wastes, thus giving, in fact, a handsome profit, in most cases in common practice.

The jacket must supply, for best effect, as assumed in the Rankine cycle, the difference between the quantities of heat demanded in the case of the non conducting cylinder and its parallel cycle employing dry and saturated steam throughout the

stroke. This difference is given in the table, and shown graphically in the accompanying plate, where the jacket is assumed to supply just sufficient heat to hold the charge in the dry and saturated state. Were it required that the expansion curve should be an equilateral hyperbola, it would be necessary that the jacket should supply still more heat, and that the working steam should become more and more superheated, as expansion progresses. For this case, Cotterill has computed the following quantities: *

IDEAL CONDENSING ENGINE, JACKETED.

STEAM PRESSURE, 95 POUNDS, ABSOLUTE; BACK PRESSURE, 3 POUNDS.

r	Work Per lb. B.T.U.	Lbs. Steam per I.H.P. Per Hour.			Jacket Steam.	Heat Per I.H.P. Per Min. B.T.U.	Efficiency.	
		Charge.	Jacket.	Total.			Jacketed.	Not Jacketed.†
1	79.0	32.5	0.	32.5	0.000	594	0.072	0.055
2	126.0	20.3	0.73	21.0	0.035	382	0.112	0.085
5	180.1	14.2	1.15	15.3	0.075	280	0.153	0.118
9	206.7	12.4	1.33	13.7	0.091	250	0.171	0.134
13	218.0	11.8	1.44	13.2	0.110	242	0.177	0.139
Complete	228.0	11.2	1.77	13.0	0.136	236	0.181	0.141

In this case, the jackets must supply about 10% of all steam used, for ratios of expansion usual as maxima in single cylinders, and about half this proportion for each of the elements of the multiple-expansion engine. The jacketed engine is here, as in the preceding ideal case, less efficient, by between 1 and 2%, than the ideal engine with non-conducting cylinder; the difference being a nearly constant quantity for the range of expansion usual in any one cylinder. The last column shows the efficiency of the corresponding cylinder when an exhaust-waste of 20% is assumed.‡ This, as is seen, gives a nearly constant comparative gain by the jacket of about 4% efficiency and of 25% on the heat, steam, and fuel accounts. As a matter of fact, however, the exhaust-waste would be greater at high ratios of expansion, and the gain by the jacket, as elsewhere abundantly illustrated, considerably greater than here indicated.

* *The Steam-Engine*, 2d edition, 1890, p. 51 *et seq.*

† Exhaust-waste assumed constant at 0.20.

‡ Cotterill, p. 55.

THE REAL ENGINE differs from the ideals which have been discussed in the fact that it is subject to three characteristic forms of thermal and dynamic waste, amounting usually, as a minimum, to perhaps 30% of the total heat-supply:

(1) External heat-wastes, by conduction and radiation of heat to adjacent bodies; not much exceeding, usually, 1% in good practice.

(2) Internal heat-wastes of heat by transfer from the working steam, early in the cycle, to the metal surfaces of the cylinder walls, and to any water present in the cylinder or its passages, with subsequent return of this heat to the working fluid, too late in the cycle to have thermo-dynamic value; amounting, as a minimum, to about 20%.

(3) Dynamic wastes by friction, causing the re-transformation of a part of the work of the thermo-dynamic cycle into heat, and its dispersion as waste; this seldom amounts to less than 10%, in the best of condensing engines, and about one-half that figure for non-condensing engines.

The real engine is subject to the same thermo-dynamic and unavoidable waste of rejected heat as the ideal having the same cycle. This, together with the two thermal wastes and the one dynamic waste of energy, above noted, constitute the total of the wastes of all engines; and the sum of the extra thermo-dynamic wastes, in good practice, on a large scale, even, is usually fully equal to the essential, unevadable, thermo-dynamic waste of the corresponding ideal engine. The progress of the steam-engine during the century has been, largely, that of reduction of these characteristic wastes of the real engine.

The distribution of utilized and wasted energies in a compound engine of moderate pressure and expansion, as reported by Hirn, is seen below: *

Heat supplied per Stroke, in saturated steam	B.T.U.		Per Cent.	B.T.U.
		Work	13.22	95.17
	686.27	Radiation	1.38	9.92
Superheat	33.61	Condensed steam	4.83	34.81
		Condensing water	79.84	574.76
Total	719.88	Total	99.27	714.66
		Error	.73	5.22
			100.	719.88

* *Bulletin de la Société de Mulhouse*, 1877.

The balance is remarkably close, when the time of the investigation and the facilities of the time are considered.

Two of Mr. Mair's tests of engines, the one (A) simple, the other (B) compound, as reported to the British Institution of Civil Engineers, give the following final results.* They were of, respectively, 101.79 and 127.4 I.H.P., at 14.6 and 24 revolutions per minute, employing steam containing, initially, 4% water.

RESULTS OF TRIALS.

	A	B
Steam supply to jacket, per cent	4.0	13.20
Loss by back-pressure, "	11.5	7.9
Initial condensation, "	43.8	22.3
B.T.U. per I. H. P. per minute	429	291
Lbs. dry steam per I.H.P. per hour	19.5	13.7
Max. ideal efficiency	0.28	0.30
Actual efficiency	0.10	0.147
B.T.U. in injection-water per I.H.P. per hour	.....	220

The steam-jacket affects and is affected by all these forms of waste and their variations of quantity. By its enlargement of the external surface of the cylinder, and its considerably higher mean temperature, it increases in a sensible proportion the radiation of heat from the exterior; while this exaggeration of waste is reflected in its own slightly increased weight of water of condensation, which necessarily includes all due to transmission of heat outward as well as inward. It has for its purpose the reduction of internal heat-waste, and its action is reciprocally modified by differences in quality of steam supplied, and in quality as determined by the relative weight of charge and extent and normal condensing power of the sides. The dryer the steam furnished the engine, and the less the condensing effect of the walls, the better the resultant effect of jacket-action and other conditions tending toward increased efficiency. Under favorable conditions for its effective action, the jacket should produce a saving of several times its own weight of steam supply, of at least three or four times in simple engine, and two to three times its weight in multiple-cylinder engines. The deflection of 5% of the steam supply into the jacket is often found to

* "Independent Engine-testing;" *Proceedings Institution of Civil Engineers*, Vols. LXX., LXXIX.; 1881-1882, 1884-1885.

effect an economy of 15%, as compared with the same engine unjacketed. Even the friction of the engine has some relation to jacket-action. The result of friction of piston and cylinder is the transformation of dynamic into thermal energy, producing, by warming the rubbing surfaces, precisely that result which the jacket is intended to effect, and, to that extent, re-enforcing the action of the latter. On the other hand, the jacket, keeping the cylinder in shape, reduces the friction of the machine, and its effect on lubrication may have a similar or an opposite result.

THE EXTERNAL THERMAL WASTES of the steam-engine vary with the difference of temperature of the steam-cylinder and the surrounding atmosphere, the thoroughness of protection by non-conducting coverings or by polish of surfaces exposed to the air, and the extent of surface acting to produce waste per unit of weight of steam used in the unit of time. This waste should always be small, and may usually be made insignificant. The higher the steam-pressure, the lower the ratio of expansion, the better the finish of polished parts, and the efficiency of the "lagging" of covered portions of the cylinder; and the larger the engine and the greater its power per cubic foot of volume of cylinder, and, also, the warmer the engine-room or the location of the engine, the less this loss. The method of attachment of cylinder to frame, even, will affect its magnitude; since the larger the surfaces of contact the better the conduction of heat to the greater mass. Its total amount varies, in the best of practice, from 1% in large simple engines to 10% in small multiple-cylinder engines.

In the unjacketed engine this waste is a drain upon the heat of the entering charge of steam, causing an internal condensation of corresponding amount. That is to say, an engine radiating 1% of its heat-supply into space is subject to at least 1% more internal condensation than one covered by perfectly non-conducting lagging, and loses by that process correspondingly less than the badly protected engine, radiating and conducting much of its heat to surrounding objects. In jacketed engines this waste is supplied wholly from the heat furnished the jacket, and its amount is the measure of a similar variation of quantity of jacket-water, while the cylinder condensation becomes independent of efficiency of external covering. External thermal waste and a steam-jacket have precisely oppo-

site effects on internal wastes. The former, drawing for its heat upon the jacket, however, has no influence upon true jacket-action, which proceeds precisely the same, whether external wastes are small or great. The drain of heat into the cylinder is measured in the jacket-water, nevertheless, only after the portion due external waste has been separated, while the effect of the jacket is measured by the sum of these two distinctly different results of its action—the removal of the injurious effect of external loss, as affecting internal condensation, and the true jacket-action, due to the passage of heat from the jacket into the cylinder. The former is the same in effect as a perfectly non-conducting covering; the latter is a positive heat-supply, introducing a minor to check a major waste. An engine condensing but 1% of its steam-supply in its jackets illustrates the first case; one condensing, under similar conditions, otherwise, 10%, illustrates the second. The first jacket acts simply as a lagging, so far as it affects the internal working of the steam; the second actively supplies heat to check internal wastes. In multiple-cylinder engines the larger the number of cylinders, and the greater their aggregate area for a stated power, the greater this loss and the lower the total efficiency. The total internal wastes of the series may, in this case, be taken as that of a single cylinder; but external waste is the sum of the losses from all. The value of the jacket is measured, as already noted, by the excess of its effect in reducing initial condensation over the amount of jacket condensation due the two effects here referred to. Should the latter sum be the greater, the jacket is a source of waste instead of gain.

The magnitude of this waste may be reckoned as from 2 B.T.U., with rough cast iron, to 0.5 B.T.U. per square foot of well-covered cylinder surface, per degree range of temperature between internal and external walls, and per hour. In the most economical engine of which we have record, the triple-expansion pumping engine of the city of Milwaukee, this figure is 0.7 B.T.U. For exceptionally well-covered cylinders this quantity may be reduced one-half. For engines of about 500 H. P. this amounts to from 1 to 2% of all heat supplied, in the best practice. Mr. J. S. Mair reports from 1 to 3%, usually less than 2%. Where comparing engines of different sizes, it may be roughly estimated as varying inversely with the size of cylinder, and taken as 100% divided by the diameter of

piston.* With multiple-cylinder engines, well covered, it has been reduced to about this latter figure, the diameter of the low-pressure piston being taken as the divisor. In a series of agricultural engine trials, in Great Britain, reported by Messrs. Bramwell and Anderson, some years ago, the percentages of radiation losses were substantially the same with simple and compound engines of equal size and similar class in design and workmanship.† Great variation was observed among the various engines tested; the loss by external radiation ranging, for boiler and engine together, from 5 to 15% of the heat generated in the furnace. Of this waste a small fraction, possibly one-tenth to one-twentieth, for small double engines and large single engines, respectively, is to be charged to the surfaces of the engine cylinders. Comparing simple and compound engines on the basis of power delivered, the latter were found to dissipate 2,000 to 5,000 B.T.U. per hour per brake horse-power; while the former wasted from about 2,500 to above 6,000 B.T.U., in all cases reckoned as from engine and boiler together.

Experiments of Dr. A. M. Mayer, communicated to the author, give the following: The relative radiations from a cube of cast iron, having faces rough, as from the foundry, planed, "drawfiled," and polished, and from the same surfaces oiled, are as below:

Surface.	Oiled.	Dry.
Rough	100	100
Planed	60	32
Drawfiled	49	20
Polished	45	18

The singular fact here appears that the oiling of smoothly

* Dividing 20,000 by the product of diameter of cylinder in inches—of the large cylinder, if multiple—and the range of temperature on the Fahrenheit scale between that of the steam and that of the surrounding atmosphere, the quotient may be taken as a rough measure of the percentage of heat wasted externally by the best clothed engines, *i.e.*,

$$h = \frac{20{,}000}{d\ \Delta\ T}.$$

When, as is not uncommon, ΔT is about 200°, this becomes

$$h = \frac{100}{d};$$

and corresponds closely with the best recorded cases. The waste is often 50% higher.

† *Manual of the Steam-Engine*, § 126, pp. 484–6.

polished castings, as of cylinder-heads of steam-engines, more than doubles the loss of heat by radiation, while it does not seriously effect rough castings. On the other hand, the experiments of the author and his coadjutors show that, by superficial oxidation, the heat transfer between metal and steam in the engine cylinder may be reduced to about two-thirds the usual quantity, while the addition of a non-conducting varnish may make this figure as low as about one-third.*

THE INTERNAL THERMAL WASTES of the engine are always of serious amount, and often constitute a vastly greater charge upon the machine than all others together, even including the unavoidable thermo-dynamic waste which, in the best of engines, amounts to 75% and upward of the demand in the ideal representative case; and 90% in the real case is a moderate total exhaust waste, comprehending both these losses. The following statement gives roughly the magnitude and rate of reduction of this loss in the standard forms of pumping engine, at the dates given:

Date.	Engine.	Duty.	Internal Waste. Per cent.
1750	Savery	5,000,000	95
1760	Newcomen	12,000,000	92
1775	Watt	20,000,000	80
1800	Watt	30,000,000	70
1820	Cornish	50,000,000	50
1850	Cornish	80,000,000	40
1870	Compound	100,000,000	30
1880	Compound	120,000,000	30
1890	Triple-expansion	125,000,000	25
1893	Triple-expansion	145,000,000	20

The latest of these engines, employing steam of 125 lbs. pressure, by gauge, expansion 20 times, and with a condenser pressure of about 2 lbs. per square inch above vacuum, should, internal wastes aside, do its work with about 9.25 lbs. of steam per hour per horse power. It actually requires 11.68 lbs. as the minimum record, the internal wastes being about 20% of the heat supplied. Without the jacket, the internal waste would probably be about 50% higher. Could all wastes be extinguished, the duty would become about 200,000,000, or that of the ideal representative case.

* "The Final Improvement of the Steam Engine." R. H. Thurston. *Proceedings of the United States Naval Institute*, 1891.

Internal thermal waste, initial or cylinder condensation, as it is variously called, is known to occur mainly on the instant of opening the steam-valve to admit the entering charge, and to be substantially complete, in all ordinary cases, at the time of closing of that valve. It thus is produced by the surfaces of the cylinder wall exposed to contact with prime steam up to the latter point in the stroke of the piston. This area includes the surfaces of head and piston, of the port-passages between cylinder and lower face of valve, and portion of the cylinder proper uncovered by the piston in its movement up to the point of cut-off. It is known, also, that, as a rule, the intensity of this action depends upon the range of temperature through which these surfaces pass between that of initial steam and of back-pressure. It is still further known, both by theory and by direct experiment, that the quantity of this waste varies very nearly as the square root of the time allowed for the cooling of the cylinder wall to the minimum temperature of that range. Heating to the maximum temperature occurs with enormous rapidity; but cooling is a comparatively slow process. Whatever the amount of heat rejected by the metal, it will be fully and almost instantly returned when the steam again strikes the metal. The "time of exposure," as it has come to be called, is, therefore, that of action of the exhaust, and not, as originally assumed by Rankine, for example, that of action of the entering steam.* The total area thus acting varies greatly with the form and extent of the "dead spaces" or clearances. In some Corliss engines these have but little more than 1% of the volume of the cylinder, and a total area but slightly in excess of the sum of piston and cylinder-head surfaces; while in other cases, with single valves, particularly, they amount to 10 or even 15% of the cylinder volume, and have an area double that of the two heads.† While experiments, as might be expected, give, often, discordant results, in consequence of the great variations of heat-transferring power of the surfaces of different engines, and of the same engine at different times, or with variations of qual-

* For perhaps the best proof of this fact, see *Transactions American Society Mechanical Engineers*, Vol. X., 1889, No. CCCXLIX.; where the so-called constant in the Rankine formula, based on this assumption, varies from 0.27 to 0.83, a range of between 3 and 4 to 1.

† Cotterill: *The Steam-Engine as Thermo-dynamic Machine*; Thurston's *Manual of the Steam-Engine*, Vol. I., Chap. V., arts. 128–130.

ity of the working fluid, the following may be taken as good and usual measures of this waste in common practice:

$$W = \frac{W a \sqrt{r t}}{d};$$

where

W = weight of steam initially condensed;
W' = weight demanded for the corresponding ideal case;
r = ratio of expansion;
d = diameter of piston, in inches;
t = time of one revolution, in seconds.

The coefficient a ranges from 3 to 4 in ordinary examples, for unjacketed cylinders, and from 2 to 3 for well-jacketed engines. Occasionally, with exceptionally dry steam, or with unusual "reluctance" of heat-transfer, or perhaps with uncommonly slight precipitation of water on the metal, these coefficients may fall to one-half the maximum figure just given.* With well-superheated steam a falls, in some cases at least, to unity.

The weight of steam condensed on known areas of cylinder wall may be taken as, on the average, approximately per stroke,

$$W = c A (T_1 - T_2) \sqrt{t};$$

where

A = area in action, in square feet;
T_1 and T_2 = temperatures, initial and final.
c = a constant coefficient having the value, for similar conditions to those above, of, approximately, 0.0025 for time, t, occupied by one revolution, taken in minutes, or 0.00033 for time in seconds. Measuring the heat-waste in B. T. U., the constant becomes $c = 2.5$ for minutes, and $c = 0.33$ for seconds per revolution, or time of complete cycle.

These figures correspond, in the common forms of mill engine, to a percentage of steam supplied very generally nearly equal to $0.2 \sqrt{r}$, in engines of one or two hundred horse-power and at customary "moderate" speeds of piston. A not unusual proportion of steam condensed is 20 to 30%. The computations of probable values can never be taken as more than approximate;

* *Transactions American Society Mechanical Engineers*, Vol. XI., 1890, pp. 170, 175, tables.

as every variation of quality of steam supplied, of conditions of condensing surfaces, or of quantity of water adhering to the wall, or collecting in the engine, or of any of the essential working conditions of the machine, modifies the rate of heat-transfer between the steam and the metal. The approximation may be made fairly close and satisfactory, however, for most cases in practice. It is far less easy to predict the efficiency of the steam-jacket in reducing this waste; as the facts of experience exhibit enormous and often unaccountable variations of jacket-action, under apparently identical or very similar conditions; while they show clearly, nevertheless, that a well-made and well-maintained jacketing will always, at high ratios of expansion and at low to medium engine speeds, pay for itself handsomely. The results of experiments at Sandy Hook, Conn., are seen in the table given herewith.*

CYLINDER CONDENSATION.

SIMPLE NON-CONDENSING ENGINE.

(*Experiments at Sandy Hook,* 1884.)

Reference Number of Experiment.	Duration of Experiment.	Stroke of Cylinder, in feet.	Diameter of Cylinder, in feet.	Reciprocal of True Number of Expansions.	Number of Strokes per minute.	Quality of Steam used (British Units).	Absolute Pressure in lbs. per square inch at Cut-off.	Absolute Steam-pressure of Exhaust at Mid-stroke, lbs. per square inch.	Temperature of Cylinder at Cut-off (Fahr.).	Temperature of Cylinder for Exhaust.	Specific Volume of Saturated Steam at Cut-off.	Ratio of Actual to Indicated Steam at Cut-off.	Condens. per sq. ft., per min., per 1° F. In lbs. of Steam.	Condens. per sq. ft., per min., per 1° F. In British Units of Heat.
	h. m.													
(7)	1.40	3.5	1.5	**0.589**	136.52	Dry.Sat.	61.54	4.22	294.2	155.2	430	1.294	0.01576	**14.29**
(8)	2.00	3.5	1.5	**0.443**	135.9	" "	68.34	3.91	307.14	152.15	390.13	1.371	0.01929	**18.00**
(9)	1.55	3.5	1.5	**0.330**	134.64	" "	62.10	4.48	294.80	157.56	426.35	1.512	0.01909	**17.31**
(10)	2.00	3.5	1.5	**0.131**	137.9	" "	49.11	3.65	279.84	149.4	530.9	2.003	0.01652	**15.15**
(11)	2.00	3.5	1.5	0.208	138.03	" "	**78.80**	3.24	311.02	144.58	342.2	1.544	0.01544	**13.82**
(12)	1.45	3.5	1.5	0.206	141.44	" "	**66.89**	3.83	299.69	151.88	398.1	(?)1.917	0.02563	(?)
(13)	2.00	3.5	1.5	0.244	143.46	" "	**53.21**	3.24	284.95	144.58	492.1	1.583	0.01416	**12.94**
(14)	2.00	3.5	1.5	0.210	137.82	" "	**39.83**	3.61	266.74	149.02	642.9	1.707	0.01528	**14.16**
(15)	2.00	3.5	1.5	0.242	135.85	" "	**26.74**	3.46	244.26	147.21	936.6	1.700	0.01372	**12.94**
(16)	3.00	3.5	1.5	0.412	135.96	" "	**65.36**	14.70	298.16	212	407.5	(?)1.122	0.00884	(?)
(17)	2.30	3.5	1.5	0.420	137.14	" "	**50.42**	14.82	281.30	212.4	504.5	(?)1.307	0.02288	(?)
(18)	3.00	3.5	1.5	0.401	135.02	" "	**40.52**	14.88	267.87	212.6	620.8	1.190	0.01416	**13.11**
(19)	3.00	3.5	1.5	0.466	133.04	" "	**28.40**	14.84	247.56	212.49	886.6	(?)1.376	0.04090	(?)
(20)	1.30	3.5	1.5	0.938	**125.95**	" "	27.38	3.15	245.56	146.31	916.2	1.322	0.01404	**12.23**
(21)	2.00	3.5	1.5	0.961	**100.60**	" "	28.35	3.86	247.46	151.57	886.8	1.403	0.01528	**14.36**
(22)	1.45	3.5	1.5	0.981	**67.48**	" "	28.53	4.06	247.81	161.94	881.6	1.504	0.01449	**13.62**
													Average..	**14.33**

* "Marks on the Steam-engine;" Thurston's *Manual of the Steam-Engine,* Vol. I., p. 509.

Numbers 7–10 illustrate varying expansion; 11–15, varying steam-pressures, condensing; 16–19, varying pressure, non-condensing; 20–22, varying speed of engine.

Cotterill deduces, for the fraction of steam-supply condensed initially, the following expression, which gives substantially the same results as that of the writer; the same functions of time and of dimensions of engine being employed, and the logarithm of the ratio of expansion being substituted for the square root of that ratio:

$$x = \frac{C \log_e r}{d \sqrt{N}};$$

in which d is the diameter of cylinder in feet; N is the number of revolutions per minute; and C is a constant which may have values from 3.5 to 7.5, according to quality of steam and condition of the cylinder wall in respect to heat-transfer. The average of all cases reported is about 5.5; the figure being, for the Sandy Hook experiments reported by the writer, 5.3; that investigation representing a fair average case. The lowest values of C correspond with most favorable economical conditions; the highest values indicate either wet steam, standing water in the cylinder, or peculiarly rough walls or otherwise high-conducting and heat-transferring power in the surfaces of the cylinder exposed during the period of initial condensation.

The quantity of heat thus abstracted from the entering steam and absorbed by the cylinder wall amounts to from 500 to 700 B.T.U. per square foot of that surface per minute, at low expansions, increasing with increasing expansion. A rough estimate may be taken as

$$H = \frac{600}{\sqrt{r}};$$

where H is the quantity of heat thus transferred, in the cases above referred to. This corresponds very nearly to

$$H = 3 \cdot \Delta T \cdot \sqrt{r};$$

where ΔT is the range of temperature between initial and exhaust pressures. The total amount transferred ranges from

1,000 to 3,000 B.T.U. per square foot per minute, in common cases.

The condition of the surfaces probably exerts a very important influence upon the magnitudes of these quantities and the economical performance of the engine. A smooth, greasy, cylinder wall, especially in presence of dry steam, permits but little heat to be transferred; and the writer has shown the possibility of considerably decreasing this waste by saturating a deeply oxidized surface with the resin from a drying oil, or with other good non-conductor.* The rush of the entering steam was relied upon by Willans to sweep off the precipitated water from the cylinder wall in his engines, and thus to aid in preserving dry surfaces.†

A careful examination of all the evidence to date indicates that it may usually be assumed, without important error, that initial condensation is practically complete very early in the stroke, and that it is substantially all produced by the surfaces of the clearance and port-spaces, and therefore, in any one engine running at regular speed and only varying in degree of expansion, a nearly constant quantity.‡ The varying ratios which have been assigned it, as a fraction of the steam consumed by the engine, in such cases come of the variations of the amount of steam supplied, simply. With constant condensation, range of temperature and of pressure, and speed of engine constant, it becomes easy to compute the total expenditure of heat, steam, and fuel, by ascertaining that required for the ideal case, and adding a constant quantity for internal wastes. The quantity so taken may usually be assumed to be about 15 lbs. per square foot of internal area of cylinder for non-condensing engines, and 20 lbs. for condensing engines, in which the range of temperature is commonly nearly twice as great.

The discharge of heat from the internal surfaces of the engine-cylinder must obviously depend both upon the physical

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* *Proceedings United States Naval Institute*, 1891.

† Probably one reason of the extraordinary economy of the Milwaukee triple-expansion pumping engine may be the fact that its surfaces exposed to this action are of small area, and that the steam has its exit through the cylinder head, and thus exerts no cooling action on extended areas of condensing port surface.

‡ This fact is, perhaps, one of the best possible proofs of the importance of applying steam-jackets, where used, to the heads, and, if practicable, to the piston, rather than to the barrel of the steam-cylinder.

condition of the metal and upon the nature and extent of the surface coating of moisture, grease, or other adherent substance, with which they may be covered in regular working. The regularity or variability of this deposit of covering, or of this condition of surface, will also affect both the average action of the engine and its uniformity and reliability of economical performance.

The same variability of heat-transfer which has been noted with external surfaces undoubtedly occurs inside the engine with differences of surface condition. Presumably, the radiation effect may be taken in presence of moist steam, for dry and rough surfaces, as not far from 2 B.T.U. as a maximum per square foot per hour per unit difference of temperature on the Fahrenheit scale. For smooth surfaces about 0.5 B.T.U., and, if the latter are bathed in oil, about one thermal unit. Where the surfaces are hot and dry and polished, and in presence of hot, dry steam, it is supposed that the discharge of heat will not exceed one-tenth the amount transferred from the metal when rough and exposed to air. The figure for this case may be taken as about 0.05 B.T.U. The rate of delivery of heat to the metal to and from a mass of steam evaporating from or condensing upon it, on the other hand, is, as has just been seen, vastly greater than that of the reverse transfer. It is, however, the rate and quantity of loss of heat from the metal during the period of cooling only, during expansion and exhaust periods, that it is important to be able to compute.

THE RATE OF HEAT-INTERCHANGE between steam and cylinder wall is dependent upon the physical qualities of both the fluid and the metal. The readiness with which the former yields its heat of vaporization in condensing, and accepts heat in re-evaporating; the conductivity, density, and specific heat of both metal and fluid, and the character, especially, of the surfaces across which the transfer of heat takes place, are the principal determining conditions.

Water condenses upon a cold surface, by surrender of its latent heat as steam, instantly upon contact with the solid, and its rate of condensation is only limited, apparently, by the rate of flow into the substance to which it is transferred, of the heat thus surrendered. The progressive reduction of rate of condensation observed, where sensible, is due to the rise in temperature of that surface, with corresponding loss of the tem-

perature-head; to which head that rate of transfer is proportional. Could the area upon which condensation takes place be held down to its initial temperature, and the heat-flow thus made such as to carry away the heat received as rapidly as yielded by the vapor, the steam would condense with a rapidity only limited by the obstruction of heat-flow from it to the metal, through the action of the accumulating cushion of water of condensation, or by the limit of velocity of flow of the fluid toward the condensing surface.

The metal takes up the heat from the steam partly by storage, a process proportional in effectiveness to its specific heat, and partly by conduction to remote portions of the solid, a process effective in proportion to the conductivity of the substance, and its density. The cylinder wall is usually composed of cast iron, having a density of about 450 lbs. per cubic foot, a specific gravity not far from 7.25. Steel "liners" are used on marine-engine cylinders, having a density of 480 lbs. per cubic foot, a specific gravity of about 7.75. The specific heat of cast iron is variable with composition, and is usually taken as one-ninth, water being unity. Regnault gives 0.118 for wrought iron or steel, and 0.138 for dark cast iron, 0.12728 for white iron. One-eighth is perhaps a fair approximate figure for present purposes. The conductivity of solid iron, according to Forbes, is sufficient to transmit 7.5 B.T.U. per minute, or 450 B.T.U. per hour, across a section one foot in area, with a temperature-head of one degree Fahrenheit.

One cubic foot of iron thus stores or yields 60 B.T.U. for each Fahrenheit degree variation of temperature. One cubic foot of water, at its temperature of maximum density, stores or re-stores 62.425 B.T.U. per degree. Its density decreases, however, at the rate of about one-tenth of a pound per cubic foot, for an increase of 50° in temperature, and becomes, at 335° Fahr., corresponding to a gauge-pressure of 95 lbs. per square inch, 56 lbs. per cubic foot. At the now customary pressures and temperatures of steam, the water of condensation on the cylinder wall stores about 55 B.T.U. per cubic foot per degree. It thus stores about 10% less than iron in the same volume. Each pound of steam, condensing, surrenders to the iron, or, in evaporating, receives from the metal, the latent heat of vaporization—which, at usual pressures in good practice, is something less than 900 B.T.U.—and the small portion of

sensible heat corresponding to the minute fall of pressure occurring at the moment. It will be here assumed that 900 B.T.U. per pound is thus given up to the cylinder wall, or received from it, in transfers between it and the steam.

The rate of transfer in iron is expressed by the equation

$$Q = \frac{a A t \Delta T}{s}, \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (1)$$

where A is the area affected, and s the thickness of the wall or the space traversed between the surfaces or planes taken as boundaries, and ΔT is the difference of temperature observed or assumed. In the mass of solid iron, according to Forbes, the coefficient a is 7.5 for time, t, in minutes, 450 for t, in hours, and for s in inches, and T in Fahrenheit degrees. We shall find this coefficient extremely variable, with variation in the surface conditions affecting transfer, in such cases as are now to be studied.

The most important difference usually comes of the fact that the resistance to transfer of heat across a thin plate, as in the steam-boiler or in a condenser, is often due mainly to the surface layer of non-conducting material, which always consists, in part, at least, of oxide, and, in the boiler, with a covering of soot on the one side and of incrustation or sediment on the other. Thus, taking 5,000 B.T.U. per hour per square foot of heating surface as a common mean rate of transfer in the steam-boiler, and the difference of temperature producing flow as averaging 1,000° Fahr., the coefficient is found to be, for an average thickness of a quarter inch,

$$a = \frac{Q\,s}{\Delta T} = \frac{5{,}000 \times 0.25}{1{,}000} = 1.25 \quad . \quad . \quad . \quad . \quad (2)$$

one-sixth the rate obtained as above, for transfer through the metal.

Except for this surface resistance, the rate of transfer would be, according to Forbes's experiments,

$$Q = \frac{450 \times 1{,}000}{0.25} = 1{,}800{,}000 \text{ B.T.U.},$$

per square foot per hour, or the equivalent of an evaporation ot about a ton of water into steam on each square foot each hour.

This is three hundred times the usual maximum. Isherwood found cast-iron condenser tubes to transmit 316 B.T.U. per hour per square foot per degree, irrespective of thickness up to the limit of his experiments—0.375 inch—and wrought iron gave 20% higher figures. Copper and brass transmitted about double the figures for iron, under similar conditions.* The steam-engine cylinder, as a condenser, in the initial portion of the stroke usually has fully as high, and often much higher, effectiveness.

Thus we find the following, according to Cotterill, as the heat-transfer in the engine cylinder in Emery's experiments on the *Bache* and the *Dallas*: †

INITIAL CONDENSATION—EMERY.

Engine and Dimensions.	r	N	A per lb.	Cond. per cent.	Heat-waste: B. T. U. per lb.	per sq. ft.	per sq. ft. per min.
Bache.	12.6	40	49.2	62.8	558	11.8	2,736
25″ × 2′	8.6	46	48.2	47.4	421	8.8	2,112
$p_1 = 90$	5.1	54	47.4	30.3	269	6.7	1,608
Dallas.	5.1	49	60.0	34.5	317	5.3	1,272
36″ × 2¼′	3.9	57	53.6	26.4	243	4.5	1,080
$p_1 = 47$	2.9	64	46.6	21.5	198	4.2	1,008
					Average.....		1,636

The average is 1,636 B.T.U. per minute, or 98,160 per hour, corresponding to a condensation of about 1.8 and 108 lbs. of steam, respectively.

These quantities of steam condensed amount to from 16 to 65 B.T.U. absorbed by the cylinder wall per square foot per revolution. The time actually occupied in the condensation of the charge is usually less than one-fourth of one second.

Assuming a penetration of one-tenth of an inch to the layer of sensibly constant mean temperature, the above average per minute gives as a mean,

$$\Delta T = \frac{1{,}600\ X \times 0.1}{7.5} = 21.33^\circ \text{ Fahr.}, \quad . \quad . \quad . \quad (2)$$

as the temperature-head producing flow, and the measure of the fluctuation; but the actual would be twice as great, at least,

* See *Manual*, Vol. II., p. 137.

as the average fluctuation of temperature in this superficial film. Dixwell deduced from his experiments a range of 30° Fahr. for a smaller and faster engine. For the most rapid flow in the above table—corresponding to a ratio of expansions of 12.6, instead of, as in this case, 5.1—the mean range of fluctuation would be 40° Fahr. nearly. For the minimum flow, 1,000 *B. T. U.* per minute, the mean range would seem to be about 14°. The extreme fluctuations would be, above, above 45°, 80°, and 20° Fahr, respectively. The above corresponds fairly with Forbes' figure for cast-iron.

Thus, if we take Dixwell's estimate of temperature fluctuation, and Churchill's measure of depth to the film of insensible variation, 0.1 inch, we find the mean coefficient to be

$$a = 1{,}600 \times 0.1 / 30 = 5.3, \quad . \quad . \quad . \quad . \quad . \quad (4)$$

for the average case, for one minute, 18 *B. T. U.* per hour. This figure is substantially the same as that obtained by Isherwood, whose condenser tubes had not far from the same mean

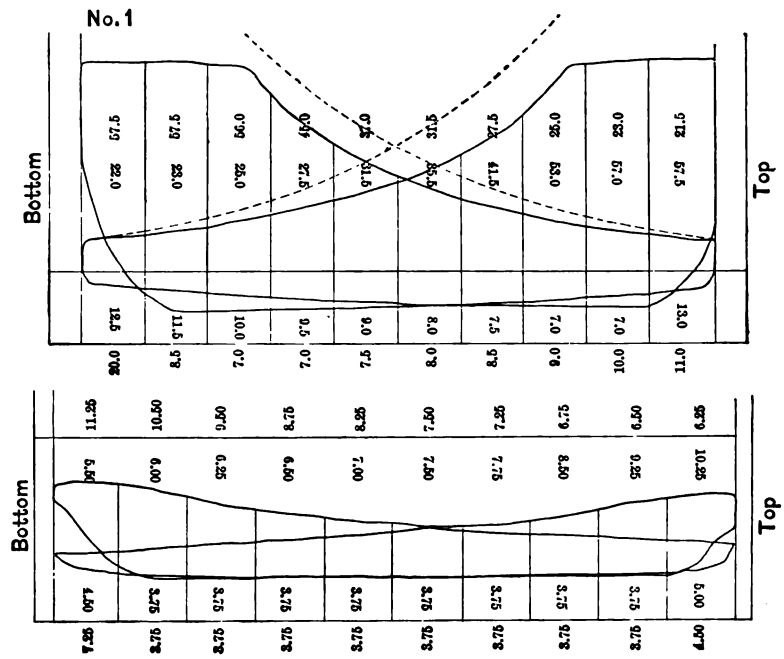

FIG. 244.

thickness of the same larger value. It is further seen that the engine-cylinder transfers heat several hundred times as rapidly as the steam-boiler.

The flow of heat through a layer of water is comparatively slow, since the conductivity of the liquid is very low, and about one one-thousandth that of iron. A film of water on the surface of the cylinder wall will thus retard the passage of heat into or from the metal, and, by alternate condensation and re-evaporation, it will itself store and restore the heat of the steam.

THE THICKNESS OF THE FILM OF METAL of a cylinder-wall affected by the rapid variations of temperatures accompanying the changing steam-pressures in the course of the cycle of the engine is known to be very slight in even the unjacketed engine; and its depth is restricted by the action of the jacket. The first recorded experiments directly determining this fact were probably those of Mr. A. A. Wilson, of the Quintard Iron Works, New York City, who, in 1882, conducted a series of interesting experiments upon one of the pumping engines of the

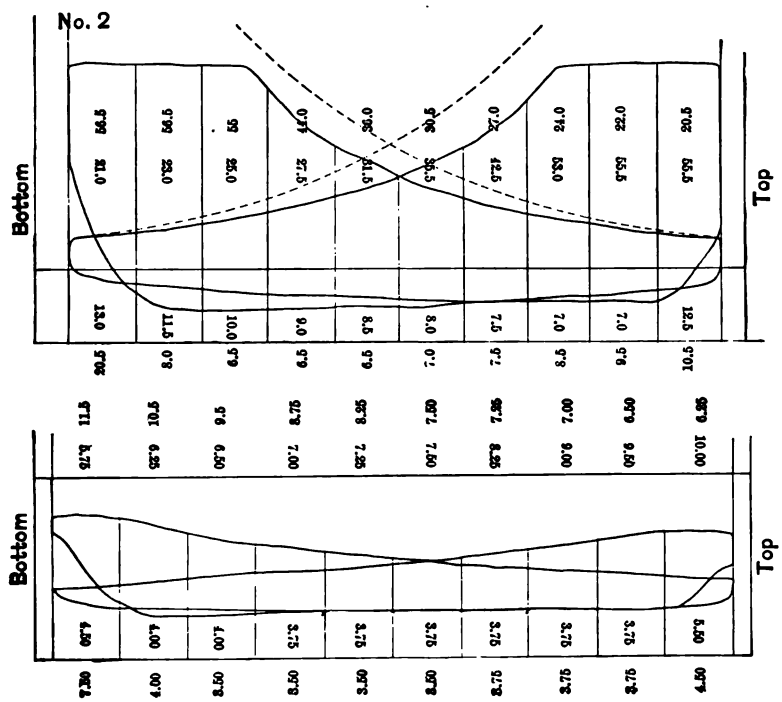

FIG. 245.

West Side Stations of the city of Chicago. The engine had the following dimensions:

Diameter of high pressure cylinder, 48 inches; stroke of piston, 6 feet; clearance, 5.15%; low pressure cylinder, 76 inches diameter, 10 feet stroke, clearance 2.5%.

Figs. 244 and 245 exhibit the forms of the indicator diagrams obtained, and the following are their principal elements and data:

	High Pressure.	Low Pressure.	High Pressure.	Low Pressure.
Card Number	1	1	2	2
Time	P.M., 3.15	3.15	P.M., 3.15	3.15
Boiler Steam	Gauge, 45	45	45	45
Recorder Steam	Abs.			
Vacuum	Lbs., 13½	13½	13.5	13.5
Revolutions	10	10	10	10
Temperature of Cylinder	273°		272°	
M.E.P., Top	27.8	3.65	27.6	3.5
M.E.P., Bottom	28.5	4.175	28.1	4.075
Average M.E.P.	28.15	3.9125	27.85	37.875
I.H.P., Top	89.235	49.68	88.59	47.64
I.H.P., Bottom	93.765	56.83	92.43	55.47
I.H.P., Total	183.0	106.52	181.02	103.11
	289.52		284.13	

The purpose of taking these diagrams was to ascertain the condition of the working charge of steam and its variations of pressure and temperature during the movement of the piston, and while the changes of temperature of cylinder-wall were being observed. It will be especially noted that the speed of engine was but ten revolutions per minute, and that the time allowed by variation of that temperature and the adjustment of the thermometer to its variations was exceptionally great, and the conditions thus unusually favorable to accuracy. The thermometer was inserted in the high-pressure cylinder-wall, as shown in the sketch (Fig. 246), and as near the point of cut-off as possible. The bulb was inserted to the centre of the metal wall, and the stem was protected by cotton waste from circulating air currents, or other heating or cooling actions. The thermometer was very sensitive, and its bulb was in contact with the metal. The indicated temperatures were absolutely constant throughout the period of the stroke of the engine, throughout the whole revolution, in fact; and its column only varied slowly, as the boiler steam and that in the steam-chest of the engine made their

ordinary fluctuations, following the temperature of the steam but about 20° below. The working steam was very wet, due mainly to cylinder condensation, and the re-evaporation was such as to produce 55% more vapor at the opening of the exhaust-

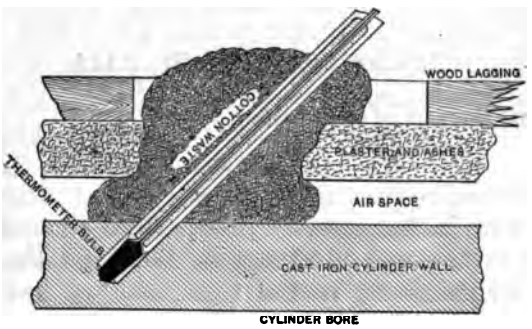

FIG. 246.

valve than at the close of the cut-off valve. The following table exhibits the pressures and temperatures observed during a period of sixteen hours:

OBSERVATIONS OF TEMPERATURES OF HIGH-PRESSURE CYLINDER.

Time.	Steam.	Temperature.	Time.	Steam.	Temperature.
6 P.M.	46 lbs.	269°	3 A.M.	47 lbs.	274°
7 "	"	"	4 "	"	275°
8 "	"	"	5 "	"	273°
9 "	47 lbs.	270°	6 "	48 lbs.	274°
10 "	44 lbs.	"	7 "	"	273°
11 "	46 lbs.	271°	8 "	46.5 lbs.	271.75°
12 "	50 lbs.	275°	9 "	46 lbs.	270°
1 A.M.	48 lbs.	274°	10 "	47.5 lbs.	272°
2 "	49 lbs.	274.5			

The outcome of the experiments was the adoption, for the engine about to be constructed, of moderately superheated steam (30° Fahr.), and jacketed heads on the lower ends of the cylinders, and the attainment of a duty when tested of above 95,000,000, a very high figure for the time. With this slight superheating Mr. Wilson reported no evidence of condensation and re-evaporation in the cylinder.

Mr. Dixwell estimates, as a deduction from his own tests, that the mean variation of cylinder-temperature does not exceed 30° Fahr. He thus takes, in the discussion of one of his engine-

trials, a case in which 920,000 lbs. of steam passed through the engine, while it made 223,000 strokes; giving 4.12 lbs. per stroke at the point of cut-off. Taking the specific heat of steam at 0.475, and of iron at 0.114, the loss of temperature of the steam which was initially superheated, having been found to be 200° Fahr.

$$4.12 \times 200 \times 0.475 = x \times 30 \times 0.114,$$

when x is the weight of iron; then

$$x = 114.4 \text{ lbs.}$$

of iron varying in temperature the specified amount, 30° Fahr. The area of surface was 56.59 square feet; and the thickness, to weigh 114.4 lbs. would be but 0.054 inch, or less than one-sixteenth.

In the course of experiments in the Sibley College laboratory, Mr. W. W. Churchill being the observer, the thickness of metal of cylinder-wall was reduced to 0.2385 inch, and the temperature of its outer service observed by means of the instantaneous action of a balanced Wheatstone bridge and a platinum-wire conductor, and without change being detected, the temperature of the entering steam being 300 Fahr. and upward, the engine non-condensing, and the revolutions 308 per minute. The load was light, however, and compression heavy. The conductor indicated a constant temperature averaging six degrees lower than that of the steam at entrance. In another series, the same general results were obtained; but the temperature recorded was less constant.

Reducing the thickness of wall to 0.115 inch, another series of trials showed slight variation of temperature and a reduction from that of the steam of about 25 degrees. Still further reducing the wall to 0.0426 inch, a fluctuation of 17 degrees became observable. These were all tentative experiments, however, and are not considered as giving reliable quantitative values.

Professor J. H. Hall has employed a plug similar to Churchill.* Using their system of temperature measurement, in a small—10-inch cylinder, 15-inch stroke—engine, the variation was found to be approximately determinable at any

* *Trans. Am. Inst. Elect. Engrs.*, May, 1891.

x = its thickness in feet;
w = weight of metal per cubic foot;
w^1 = weight of steam condensed per hour;
ΔT = range of temperature of metal;
Q = quantity of heat transferred per lb. steam;
C = specific heat of the metal;
A = area affected in square feet;
N = number of exposure per hour.

Then

$$x\,w\,A\,\Delta\,T\,C\,N = w^1\,Q; \qquad x = w^1\,Q/A\,C\,N\,\Delta\,Tw$$

If we take

$$w = 450;\ \Delta T = 50^\circ;\ C = 0.125;\ N = 3{,}000;$$

$$w' = 100;\ Q = 900;\ A = 1;$$

$$x = 0.0167 \text{ ft.};\ = 0.2 \text{ inch.}$$

The actual quantity of metal affected, measured to the layer of insensible variation, is of at least double this extent, and, probably, more nearly a half inch.

The heat yielded to the metal by the production of the water of condensation is not far from 50,000 B. T. U. per cubic foot of the liquid, at the assumed pressure; the weight of the liquid being about 56 lbs. per cubic foot and its latent heat approximately, 900 B. T. U. The weight of iron taken at 450 lbs. per cubic foot and its specific heat at 0.125, gives 56 B. T. U. per cubic foot, per degree difference of temperature. The thickness of the layer of iron which will accept the heat surrendered in the formation of one foot depth of water of condensation on a square foot is thus:

$$x = 450 \times 50 \times 0.125 / 50{,}000 = 0.056 \text{ ft.}, \quad . \quad . \quad . \quad (2)$$

or about one-twentieth the depth of the layer of water. Where the metal is affected to the depth of 0.2 inch, therefore, taken as uniformly affected, the film of water produced would be but 0.01 inch in thickness; and where, as in experiments quoted elsewhere, the depth to film of sensible variation is but 0.1 inch, and where the corresponding thickness of uniform mean variation is something less than 0.05, the film of water on the surface of the cylinder-wall would be but about 0.0025 inch., one four-

hundredth of an inch thick. Thus, as it was expressed by De Freminville, many years ago, the effect of a film of water on the surface of the metal is such that a layer resembling "the mist on a window pane" may produce all the effects observed in cases of moderate cylinder condensation, in steam-engines. Equal weights obviously, of water and iron, produce effects in proportion to their specific heats, or in the proportion of about eight to one.

Taking precipitation by condensation, of one-third of a cubic foot of steam upon the surface of the cylinder-wall one foot in area, it would produce a film of water one seven-hundredth of a foot thick, one-sixtieth of an inch. Distributed, as is more usually the fact, over several, say five square feet, the thickness of the film of moisture would be about 0 003 inch. The quantity of heat thus transferred in practice between metal and steam ranges from 500 B. T. U. per minute per square foot, with superheated steam or good jacketing, to above 3,000 B. T. U. in unjacketed cylinders with large expansion and wet steam, or from 30,000 to 180,000 B. T. U. per hour; while the surface condenser ordinarily takes up but about 15,000 B. T. U. per hour, and the steam-boiler averages 3,000 to 5,000 B. T. U. per square foot. Cylinder condensation is thus the most effective process of liquefaction known in steam engineering. The fact is due mainly to the rapid alternation of temperatures, through a moderately wide range in a very thin film of metal, or of liquid, or both; making the case one of transfer by a large temperature-head of a superficial layer, instead of, as in the other cases, by moderate head-gradients, and through considerable thickness of metal, and resistant layers of oxide, scale, moisture, slowly conducting films of water, or cushions of non-conducting steam. The thickness of water permanently covering the cylinder-wall must be, as has been seen, twenty per cent. thicker than the computed thickness of metal to produce an equal effect in heat absorption.

But the figures which have been given afford no idea of the actual rapidity of transfer of heat during the fraction of a second usually occupied by the process in each revolution of the engine. Taking the time of exposure, as sometimes observed, as one-tenth of a second, and the quantity transferred at high ratios of expansion as 50 B. T. U. per square foot per revolution, we have about one pound of steam condensed in twenty revolu-

tions, one-twentieth in a revolution; but the time actually occupied is but one-tenth of a second, and the rate is thus one-half pound per second "working time," and thirty pounds per minute, 1,800 lbs. per hour, nearly a ton an hour, and probably very nearly the full amount due the conductivity of the iron where the surfaces are clean. Taking the quantity transferred, as per low values of the ratio of expansion, as one-third the above figures, the rate of condensation becomes 600 lbs. an hour "working time" per square foot of cylinder-wall subject to the assumed change. The extreme range is probably from about 10 B. T. U. per square foot per revolution in everyday practice, affecting a depth of one three-hundredth of an inch to about 50 B. T. U., affecting a depth of one-sixtieth of an inch.

JACKET-ACTION ON INTERNAL WASTES constitutes the main point of study, experiment, and observation in the construction of this portion of the theory of the engine. The influence of the jacket on external thermal and on dynamic losses of energy is too small to demand much consideration. It affects internal thermal waste, often to a most remarkable extent, and it is exceedingly important that a way shall be found of ascertaining its precise effect, qualitatively and quantitatively. Where the steam supplied is dry and the engine-cylinder can be kept surcharged with heat, as assumed in the Rankine method of treatment of the case, the problem is capable of fairly accurate solution in the manner already illustrated; the net result being the expenditure of five to ten per cent. more heat and steam than is demanded by the ideal case, and the complete reduction of cylinder condensation. The internal wastes, in other words, are reduced from those of the unjacketed cylinder to those of the engine working steam held constantly in the dry and saturated state. For the best practice, and with large engines, this means, practically, the reduction of the internal thermal wastes to the extent of about one-half.

In most cases, however, it becomes necessary to allow for some additional expenditure of heat to meet the demand due an excess of initial heat-transfer which the most effective of usual methods of jacketing and operation cannot repress. The following table shows the results of experimental measurement of the quantity of jacket-water and of the economy produced by the jacket under widely varying conditions.

STEAM-JACKET ACTION AND ECONOMY.

(1) SIMPLE NON-CONDENSING ENGINES (B. I. M. E. REPORT.)

JACKET WATER.

No.	p_1.	Per cent.	r.	Economy, %.	Ratio.	No.	p_1.	Per cent.	r.	Economy, %.	Ratio.
2....	115	3.2	...	23.0	7 to 1	8*b*....	110	3.4	6.2	22.0	7 to 1
3*a*..	45	14.8	2.7	21.4	1.5	8*c*....	110	3.1	5.3	16.7	5.5
3*b*...	45	8.5	1.	14.9	7.7	8*d*...	110	2.0	4.4	16.7	8.
3*c*....	45	14.0	1.	11.1	0.8	8*e*....	78	1.7	5.1	14.25	8.
6....	68	3.3	3.1	10.3	3.1	8*f*...	78	1.7	5.1	11.46	7.
7*a*..	100	2.3	3.4	9.3	4.	8*g*...	78	1.2	3.8	11.36	9.5
7*b*...	100	2.3	3.3	7.25	3.	8*h*...	78	1.3	3.1	9.23	7.
7*c*....	100	2.7	3.4	16.3	6.	8*i*....	50	1.5	3.9	7.21	5.
7*d*...	50	3.5	3.4	15.9	4.5	8*j*...	50	1.1	2.8	7.32	6.5
7*e*...	78	3.2	5.1	9.6	3.	8*k*...	50	0.7	1.7	4.74	6.6
7*f*....	78	3.36	5.1	22.4	6.6	8*l*....	50	0.26	1.	1.19	4.6
8*a*.	32	2.5	7.1	19.7	8.						

(2) SIMPLE CONDENSING ENGINES.

10...	54	11.75	6.5	27.5	2.4	23*d*..	89	3.1	11.3	17.7	6.
11...	38	23.35	5.3	31.2	1.3	23*e*..	89	1.2	6.8	18.8	15.7
12...	12	1.7	1.7	2.7	1.6	23*f*..	89	1.5	5.9	12.0	8.
14...	45	17.2	1.8	19.4	1.1	23*g*..	89	1.5	5.9	14.3	9.5
15...	80	3.8	10.	23.7	6	23*h*..	64	2.5	10.7	9.4	3.4
16*a*..	17	7.25	3.4	32.4	4.5	23*i*..	64	1.8	5.4	4.0	2.2
16*b*..	16	7.25	3.4	31.3	4.3	23*j*..	64	1.5	4.4	3.9	2.6
18*a*..	14	2.4	1.75	16.3	6.6	23*k*..	64	1.6	3.6	1.5	1.
18*b*..	40	5.5	4.	6.1	1.1	23*l*..	50	2.6	1.2	6.6	2.5
18*c*..	69	4.	5.5	11.5	2 9	23*m*..	50	2.3	7.6	4.5	2.
18*d*..	65	3.5	4.5	9.5	2.7	23*n*..	50	1.4	5.8	3.2	2.3
19*a*..	67	3.5	5.7	21.5	6.	23*o*..	50	1.4	4.	0.5	0.35
19*b*..	68	4.8	12.4	30.3	6.3	23*p*..	50	1.2	3.2	0.6	0.5
20...	58	3.4	4.7	15.5	4 7	23*q*..	36	1.6	4.6	3.3	2.
22. .	42	4.9	4.3	16.6	3.6	23*r*..	36	1.1	2.3	2.9	3.
23*a*..	110	2.9	10.	23.0	8.	23*s*..	36	0.6	1.7	0.6	—1. (?)
23*b*..	110	3.2	6.4	20.0	6.3	23*t*..	36	0.2	1.	8.	40. (?)
23*c*..	88	3.0	12.	16.7	5.6						

(3) COMPOUND CONDENSING ENGINES.

24...	56	5.2	4.3	23.5	4.5	31...	47	9.9	15.8	34.9	3.5
25...	41	8.9	5.5	28.6	3.2	32...	72	7 7	9.6	8.7	1.1
26...	41	17.1	11.	31.2	1.4	33...	80	6.5	7.	15.4	2.4
27. .	45	23.4	10.3	33.6	1.6						

The number in column 1 is that assigned by the committee on steam-jackets of the British Institution of Mechanical Engineers, from whose report these figures are collated. The pressure, p_1, is that shown by the steam-gauge. The ratio in the last column is that of jacket-water to saving of steam or feed-water.

A series of experiments, by Mr. J. G. Mair, on a condensing, jacketed engine, as reported to the British Institution of Civil Engineers, gave the following figures: *

* *Proceedings Institute Civil Engineers*, Vol. LXXIX., p. 331; Cotterill, p. 350.

EFFICIENCY OF JACKET.

r.	3	4	5	6
Condensation with jacket	0.25	0.35	0.45	0.60
Condensation without jacket	0.48	0.60	0.70	0.80
Difference	0.23	0.26	0.25	0.20
Ratio	0.5	0.6	0.7	0.8
Condensation in jacket	0.05	0.05	0.05	0.05
Ratio to gain	0.2	0.2	0.2	0.25

The steam pressure was 56 lbs. by gauge. The jacket is seen to have saved approximately five times the weight of steam used in it, or, otherwise, the diversion of 1 lb. of steam into the jacket effected a gain of 5 lbs. in the expenditure of steam in the engine. The average figure for the whole collection of data above corresponds with the deduction of Professor Ewing, who, in his admirable contribution to the *Encyclopedia Britannica*, says: "When a jacket is working properly it uses in a single cylinder engine 4 or 5%, and in a compound engine 10 to 12%, of the whole steam supply.* But later practice, though confirming these deductions in many cases, often gives very different proportions, and cases are reported in which the compound engine consumes 12 or 15%, and the triple-expansion engine 15 to 20%. Thus, as long since noted by Professor Cotterill, "a small quantity of heat supplied by a jacket reduces the action of the metal in a much greater proportion." †

The following are miscellaneous figures from various sources, generally illustrating recent and good practice: Compound engines have usually been found to demand about 10% of the steam supply for the jackets where they have worked effectively. A compound engine at the Memphis Electric Light Works, Tennessee, tested under direction of the author for report to him, gave, by a Hirn analysis by Mr. Boehm, 18% expenditure in jacket and radiation, probably 16% expended in the jacket. It, however, only reduced initial condensation in the high-pressure cylinder to 40%, while raising its gain to a quality of 74%—26% moisture—at the opening of the exhaust to condenser. The engine was 22 and 40 inches by 48 inches, with steam at 90 lbs., 75 revolutions, 137 I.H.P., and the expenditure of steam amounted to 20.9 lbs per H.P. per hour. The valve

* *Encyclopedia Britannica*, article, "Steam-Engine," section 91.

† Cotterill, p. 349.

motion was that of Corliss. Another engine with automatic gear, 14 and 23 inches by 22 inches, with 90 lbs. steam, 152 revolutions, 63.6 I.H.P., demanded 26.2 lbs. of steam per H.P. per hour, condensing nearly 60% of the entering steam, delivering with a quality of 55 at release, while the jackets were using about 16% of the steam supply, including loss by radiation. The compound Reynolds-Corliss engine of the Plymouth Cordage Co., on trial for acceptance, developed 999 I.H.P., and used 21.145 lbs. of steam per H.P. per hour, with a pressure of 100 lbs. and 1.4 lbs. of coal. The jackets used 10.24% of the steam supply, and initial condensation amounted to but 21%. External radiation accounted for a loss of 2.4%. The compound engine of the Namquit Mills, by the same builders, with steam at 10 lbs., on contractor's trial, used 11% of the steam supply in the jackets and consumed 13.26 lbs. per H.P. per hour. A Worthington compound, on pipe line work, is reported to have used 15.5% to 16% of the steam in its jackets, and to have exhibited a condensation of 34%, with boiler steam at 90 lbs. The Calumet & Hecla engines, "Frontenac" and "Superior," are reported by their designer to the writer as having a condensation of 7%, in each case, in the jackets, while demanding a total of, respectively, 14.4 and 14.2 lbs. per hour per H.P. Steam was carried at 100 lbs., and the engines made 50 and 43 revolutions per minute, respectively. They were both seriously underloaded. The Lawrence pumping engine condensed 9% in its jackets; and Mr. Leavitt informed the writer that his engines have very generally demanded 10% to 12% for jacket supply.

A collection of trials of Gaskill & Holly Manufacturing Company's engines, the reports of which are published by the builders, give, in tests, by Codman 9.15, by Barrus 13 to 16, by Benezette 10.32, by Gifford 5.2 (?) and by Cooley, 11.36% of steam condensed in the jackets. Mr. Reynolds reports to the writer an average of about 10% in compounds of his design. The range is 10 to 15% according to his experience, for effective jacketing, for all kinds of multiple-cylinder engines. Mr. Emery reports for the *Gallatin* about the same figures. Mair finds a condensation in the jacket of a Woolf compound of 9% and a gain by its use of 34% giving a ratio of 3.8 to one for value of steam in the jacket as compared with that used in the working charge.

Triple-expansion engines ordinarily demand about the same amount of steam in the jackets as the compounds; although it

would probably be expected, in consonance with theory, that they would use less under similar circumstances. The Milwaukee pumping engine trials reported by the writer to the American Society of Mechanical Engineers gave, where measured, a consumption of 9.25% in the jackets; the steam-pressure being 125, the ratio of expansion about 20 and the consumption of steam about 12 lbs. per horse-power per hour, in the best trial 11.678 lbs. The speed was 20 revolutions per minute. The steamer *Iona* used 5.29% of the steam in the jacket, and exhibited 26% cylinder condensation. The steamer *Meteor* took 10% in the jackets, and the cylinder condensation was 17%. The Chicago Allis Pumping Engines, indicating 1031 I.H.P. at 16 revolutions, consumed 12.13 lbs. of steam per horse-power per hour on presumably the same jacket consumption as the Milwaukee engine. The Narragansett Electric Lighting Company's Reynolds-Corliss engine, on contractor's trial, developed 516 I.H.P. at 99 revolutions per minute, steam at 125, with 12.94 lbs. of steam per horse-power per hour; the jackets using 14.33% of the steam. The radiation loss was 1.5%.

The Lea Bridge triple-expansion engines, three in number, belonging to the London water-supply system, are found, by Mr. Mair, to have a consumption of 13.28 lbs. of steam per horse-power per hour, and their jackets average a consumption of 12.28% of the total steam used by the engines. The trials of the Massachusetts Institute of Technology experimental engines reported by Professor Peabody, at 88 revolutions per minute, demand 14.65 lbs. of steam per horse-power per hour, and use 22% of the total in the jackets, the initial condensation being still 21.5%. This is, however, a comparatively small engine. Mr. Reynolds' experience with this class of engine indicates from 10 to 15% as the usual figures for jacket-water, varying considerably with relative areas of active surfaces, magnitudes of clearances, and with temperatures and pressures of steam. He usually jackets the low-pressure cylinder with steam of lower pressure than the high-pressure cylinder.

Professor Peabody's trials of the "experimental engine," under his charge, at the Massachusetts Institute of Technology, an engine of about 150 I.H.P., triple-expansion, with cylinders 9, 16, and 24 inches in diameter, 30 inches stroke of piston, with steam at about 150 lbs. pressure, furnish other useful facts. The steam demanded at various ratios of expansion, and at about 90

revolutions per minute, ranges from 13.74 lbs. per I.H.P. to 16.25; the best work being done when all jackets were in use, the worst average being made with all jackets off. The jacket had least value on the high-pressure engine, and it proved to be unimportant, where one only was used, whether it was upon the intermediate or the low pressure engine. The total steam used in the cylinders was, at full power, from 1,600 to 1,800 lbs. per hour; the jacket-steam amounting, when all jackets were in use, and receivers jacketed as well, to 450 lbs., with no steam in receivers, and at best work reported, about 300 lbs., or nearly 20% of the total supplied the engines. The best trial reported in this series gave 125.37 I. H. P. on 13.74 lbs. of steam per H. P. per hour, and with a consumption of 1,420 lbs. total, in all cylinders, and an additional 297 lbs. in the jackets, no receiver-jackets being in operation. Without jackets, the engine used 15.87 lbs. of steam per I.H.P.; the jackets, comparing similar powers reducing initial condensation to an insignificant quantity, and saving 11.6%, by the diversion of 20% of the steam supplied from the cylinder into the jackets.* And this effect is obtained at a speed of 90 revolutions per minute.

In the case of the best work which the writer had, at its date, (1891) seen reported, Professor Schröter obtained from a triple-expansion engine of 200 H.P., steam at 156 lbs. pressure, the remarkable figure of 12.7 lbs. of dry steam entering the engine per I. H. P. per hour.† This corresponds to a duty of a trifle over 162,000,000 ft.-lbs. per 100 lbs. coal at an evaporation of 10 to 1. The efficiency of machine was 88% nearly. The jackets condensed a large percentage of the steam, proving their effective working. The total, about 20%, was distributed thus: In the first cylinder-jacket, 2.2%; second and intermediate receiver, 6.4%; in third and receiver, 10.7%; the drain of heat into cylinders being greater as their mean working pressures and temperatures fell. The total jacket-condensation was about double that of the Milwaukee Pumping Engine, reported by the writer, when the latter was doing about 10% better work (11.7 lbs. per H.P. per hour), presumably partly as a consequence of the smaller clearance surfaces of the latter and more effective system of jacketing.

* *Trans. Am. Soc. M. E.*, May and November, 1892. Vols. XIII. and XIV.; Nos. CCCCLXXXIV. DII.

† *Zeits. des Ver. Deutscher Ingenieure*, VOL. XXXIV. *Lond., Eng.*, Dec. 5, 1890, p. 669.

Mr. Donkin, as the outcome of over two hundred engine-trials, finds the jacket increases the work done from 10 to 35%.

The only conclusion to be derived from this enormously irregular and unaccountable variation of results is that the steam-jacket may be made efficient under all conditions of operation of the steam engine, even at high speeds either of piston or of rotation; but that the most certain returns are to be obtained with the largest ratios of expansion in a single cylinder, and at moderate or low speeds of engine. No law can here be detected, relating economical working of the jacket to steam-pressures, expansions or speeds. Still less can any means be here found of anticipating the probable gains by use or the jacket for any individual case. It is simply evident that where the jacket can be so applied as to bring out its inherent economical quality, it will give, with wet or saturated steam, a large gain.*

The general fact seems to be, that with an expenditure, on the average, of about 5% in the jacket of simple engines, an economy of three or four times its amount may be obtained in the total steam used by the system, under favorable conditions, and that with exceptionally successful practice, the steam diverted into the jacket may produce five to eight times as much work per unit of weight as if used in the cylinder directly. The value of the jacket seems to be about the same on the condensing as on the non-condensing simple engine; but, as might be expected, the jacket-steam has lower efficiency in the multiple-expansion engine than on the simple. The more economical the engine, unjacketed, the less the economical action of the jacket. The smaller the engine, and the things equal, the more effective the jacket; and a small engine may gain by use of the jacket at speeds far above those at which the larger engine begins to lose jacket efficiency.

The best results are to be taken as representative of the value of effective jacketing; the others being simply indicative of the extent to which bad design, construction, or management may annul its beneficial action. Culling out of the table those cases in which the gain is reported at about 15% and upward, we have the following:

* Prof. C. B. Richards, reporting, as United States Commissioner, on the steam-engines exhibited at the Paris International Exposition of 1889, says: "Steam-jackets are almost universally applied to all cylinders, of either compound or non-compound engines."—*United States Reports*, Vol. III., p. 98.

BEST RESULTS OF JACKETING.

(1) SIMPLE NON-CONDENSING ENGINES.

No.	p^1	Rev.	r	Per cent of Jacket Expenditure.	Gain.	Ratio.
2	115	203		3.2	23.0	7 to 1
8*b*	110	62	6.2	3.4	22.0	7.
8*c*	110	62	5.3	3.1	16.7	5.5
8*d*	110	63	4.4	2.0	16.7	8.
7	100	97	3.4	2.7	16.3	6.
7*f*	78	93	5.1	3.3	22.4	6.6
3*a*	45	47	2.7	14.8	21.4	1.5
3*b*	45	49	1.	8.5	14.9	1.7
8*a*	32	47	7.1	2.5	19.7	8.

(2) SIMPLE CONDENSING ENGINES.

23*a*	110	60	10.	2.9	23.0	8.
23*b*	110	60	6.4	3.2	20.0	6.3
23*c*	88	59	12.	3.0	16.7	5.6
23*d*	89	58	11.3	3.1	17.7	6.
23*e*	89	59	6.8	1.2	18.8	15.7(?)
23*g*	89	60	5.9	1.5	14.3	9.5
15	80	55	10.	3.8	23.7	6.
19*a*	67	50	5.7	3.5	21.5	6.
19*b*	68	50	12.4	4.8	30.3	6.3
20	58	53	4.7	3.4	15.5	4.7
14	45	39	18.	19.2	19.4	1.1
22	43	20	4.3	4.9	16.6	3.6
11	38	60	5.3	23.5	31.2	1.3
16*a*	17	37	3.4	7.25	37.4	4.5
16*b*	16	37	3.4	7.25	31.3	4.3
18*a*	14	41	1.75	2.4	16.3	6.6

(3) COMPOUND CONDENSING ENGINES.

33	80	55	7.	6.5	15.4	2.4
24	56	235	4.3	5.2	23.5	4.5
31	47	18	15.8	9.9	34.9	3.5
27	45	39	10.3	23.4	38.6	1.6
25	41	36	5.5	8.9	28.6	3.2
26	41	32	11.	17.1	31.2	1.7

Studying the above figures, we find that, with good jacketing, it should be expected that a pound of steam employed effectively in the jacket of a simple engine, whatever the working conditions as to steam pressure, should be worth more there than in the working cylinder itself; that the value of the jacket steam, always great, is highest with the highest pressures, as a rule, increasing, also, throughout the whole range of customary expansion, as

expansion increases, and as initial pressure increases. The same general result is observed with the compound engines; but the gain is smaller in proportion to expenditure; while both expenditure and gains are proportionally large. The jacket steam is worth, even here, three times as much as the working steam, ten per cent. in the jacket saving nearly thirty per cent. in the engine. The best results are obtained from the slowest engines generally; but most of the best figures are from engines making fifty to sixty, some at ninety revolutions per minute. None, however, approach in speed the little engine of Professor Alden, expending 6.7% of its steam by way of the jacket, and at 290 revolutions per minute, gaining 14% by its use; the very first on the list has a speed of 203 revolutions. The best work of the jacket seems to be reported, usually at from six to eight expansions, upward, although large returns are often given at but three or four. The data establish sufficiently well, were such evidence needed, the fact that the heat expended in drying and heating the cylinder-wall, in anticipation of the entrance of the working charge of steam, is much less in amount than that wasted by the condensation of the latter, where such preliminary drying and heating has not been effected by the jacket-steam, and that the ratio of quantity of jacket steam to steam saved by its use should always be expected to exceed unity, and, under favorable conditions, both quantities should be large in proportion to total steam used, with corresponding gain of efficiency. Effective jacketing with engines of moderate and low speed, and with moderate or large ratios of expansion, a single cylinder should reduce the initial condensation one-half with an expenditure of one-eighth its proportion of total steam used.

THE MAXIMUM EFFICIENCY OF JACKET would naturally be expected to be found at such a ratio of expansion as would cause the jacket-action to be limited to the expansion period, so far as the drying-off of the moisture condensed on the cylinder-wall is concerned. It is known that this action results in the transfer of heat from jacket and cylinder to the evaporating moisture with enormous rapidity, and that the dry surfaces lose heat very slowly. If carried into the exhaust period, or even far into the expansion period, this evaporation means great waste. If restricted to the expansion period, the jacket performs the required task with comparatively little loss. The earlier the cylinder-wall is dried, the greater the economy. The difference

in rate of discharge of heat between the wet and dry metal is probably not less than a hundred to one. In cases elsewhere reported by the writer this maximum efficiency of jacket was well exhibited, but the subject has not yet been sufficiently well investigated to permit very extended generalizations.* In the cases there presented the following deductions were made:

"In the first case, that of the simple non-condensing Corliss engine, the heads unjacketed, the use of the jacket reduced the cylinder-wastes from about twenty-five per cent. of the ideal consumption of steam and feed water to about half that proportion for ratios of expansion approximating six; from one-third down to about one-tenth, at a ratio of five; and apparently from twenty down to ten per cent. at $r = 4.4$. The same general effect is observed throughout, with some discrepancies which may be due either to varying action of the jacket, or to errors of observation, or to both combined, the latter being the probable fact.

"In this first case, also, the jacket gives the best results with 110 lbs. of steam, when the ratio of expansion approximates six. When the steam pressure falls to eighty pounds, the best work of the jacket occurs at a ratio not far from 4.75; while, at the pressure of fifty pounds, the value of the jacket increases through the whole range of experiment, and not only so, but the curve assumes a rectilinear form, indicative of probable improvement indefinitely in the direction of increasing expansion. The highest efficiencies, however, either with or without the jacket, are found, in this case, at the lowest ratios adopted, and indicate a maximum value at about 3.25.

"Similarly studying the performance of the condensing engine, we find the best work done, whether jacketed or not, at about a ratio of expansion of ten, at a steam pressure of 110 lbs., but that the jacketed engine reduces the internal wastes from fifty per cent at highest ratios, and from one-fourth at the lowest ratios, in the case of the unjacketed engine, to five per cent., and, in some cases, possibly to within the magnitude of the errors of observation. At a pressure of ninety pounds the best ratio seems to be, under the given conditions of operation, about 6.5 when unjacketed, and 8.5 jacketed; while the lower

* "Maximum Steam-jacket Efficiency:" *Journal of the Franklin Institute*, April, 1891.

pressures still further reduce both the efficiencies and the saving effected by the jacket. The best work of the jacket, as an economizer of heat, is done, at high pressures, at a ratio of expansion of twelve or more. In all cases, it seems to be the fact, with these engines at least, that the jacket is useful beyond the ratios of maximum efficiency of fluid.

"The compound engine was operated at altogether too low a pressure to bring out the best effect of compounding; but it exhibited the same general effects noted in the working of simple engines. The effect of the jacket is less pronounced than in the simple engine, and the efficiencies of fluid vary less with variation of the ratios of expansion. It gives its best result at ratios of expansion, ranging from 7.5 to 10.5; the variations of value being very much more observable in the case in which both jackets are in use than in either of the others, and least where only the high-pressure jacket was employed."

Maximum jacket-effect is probably always found at a point beyond that of usual practice, in the direction of increasing expansion in a single cylinder, whether in the simple or the multiple-expansion engine.

With thoroughly dry steam at admission, the jacket may need to supply, and may actually supply, very little heat while still acting with efficiency.

Maximum jacket-efficiency, so far as resulting from favorable character of steam supplied, is evidently to be expected when the work demanded of the jacket is a minimum; and this will be found to occur when the steam is supplied at maximum temperature and most nearly dry. If no water is brought into the cylinder from the boiler, to be evaporated by waste of heat from the jacket, the latter has only to do its legitimate work of preparation of the cylinder-wall to receive the succeeding charge, by drying it and heating it up to the temperature of prime steam. As elsewhere expressed by the writer: *

"Could a perfectly efficient jacket be made, in the sense of being capable of instantly and fully supplying any demand, however sudden or great, for heat needed at the beginning of the stroke, on the interior of the engine, and could the steam be supplied perfectly dry initially, the vapor would remain perfectly dry throughout the stroke; none would be condensed at

* *Manual of the Steam-Engine*, Vol. I., p. 634.

the beginning, to be re-evaporated later, at the expense of heat from the jacket; and the cost would be only that of the comparatively small normal heat-waste of a dry gas; while a saving would be effected of substantially all the initial condensation that would otherwise have occurred, and at insignificant expense.

"Under such conditions, the more readily the jacket surrenders heat, the less the amount it is called upon to yield, and to waste."

Ledieu says that, to insure efficiency of jacket, it must fulfil the following conditions:

(1) Such activity must be secured as to insure that all moisture present on the walls of the cylinder at the end of expansion, shall be evaporated before the end of the exhaust-period.

(2) It should not be so active as to cause the cylinder to be completely dry before the end of expansion.*

The writer would, on the contrary, conclude that the jacket is most efficient when so active as to present a dry hot surface to entering steam, and thus prevent, in large degree, the precipitation of moisture previous to cut-off, and, *also*, sufficiently active to dry off such moisture as *is* formed at the earliest possible instant after expansion begins. No important waste can occur, later, from dry surfaces.

What must be the result of indefinite increase of temperature of steam-jacket, by increasing the pressure of its steam or otherwise, may be judged by reference to the experience of Mr. Donkin, who, in his paper, describing the trials of the gas-flame jacket, found that "after the cylinder metal had been raised to a certain temperature, the extra quantity of heat seemed to have no beneficial effect whatever."† The waste by the transfer of heat into the cylinder during the exhaust period evidently, after passing this limit, more than balanced the gain due to drying and superheating the steam in the initial stages of the cycle. The waste during condensation and exhaust more than compensated the gain by initial checking of cylindensation and by increase of work done under the expansion line. The purpose and use of the jacket being simply and solely the prevention of initial condensation in the cylinder by action of the cylinder-walls, by the

* *Machines à Feu*, § 95, p. 509.

† *Heat Expenditure in Steam-Engines*: B. Donkin, Jr. *Proc. Inst. C. E.*, Vol. XCVIII., No. 2,043.

prevention of storage of heat in the metal at that stage, to be later withdrawn wastefully, it is evident that it may happen that very little heat may answer that purpose, and the jacket may be called upon to supply correspondingly little, although by its action nevertheless producing considerable economy. The less the drain of heat from the jacket into the cylinder, this object once being effected, the better.

THE METHOD OF TRANSFER OF HEAT, in detail, between the metal and the steam was first intelligently investigated by Hirn, and the results published in his *Théorie Mécanique de la Chaleur*, in 1876.* The example there taken in illustration will still answer the purpose in our case. He finds the following data as the result of experimental investigation and measurement:

INTERNAL HEAT-TRANSFERS.

	JACKET.	
	On.	Off.
Heat absorbed by metal	0.380	0.459
Heat returned in expansion	0.317	0.173
Heat rejected, exhaust	0.050	0.273
External radiation	0.013	0.013
From steam-jacket	0.022	...
Work of expansion	0.113	0.088

The heat absorbed by the cylinder wall during admission is partly utilized by performance of work during expansion, and the rest wasted in the exhaust. The jacket, by an expenditure of 2.2% of the total steam supplied, decreases initial condensation waste 7.9%, or 20% of its own original value; the use of a pound of steam in the jacket giving a net gain of about three pounds in the engine. The work of expansion is increased by 2.5% of total supply, and this is 28% of its value without jacket-action. The heat supplied by the jacket is a comparatively insignificant quantity; but, small as it is, it raises the proportion of heat restored and doing more or less of useful work, during the expansion period, from one-third to above three-fourths of that absorbed by the cylinder-wall in taking steam; at the same time decreasing the latter quantity 20%, and correspondingly reducing

* Vol. 2, Chap. I., p. 39.

the initial waste. It decreases the initial condensation, and at the same time adds to the amount of useful work, thus giving a gain on each side of the account, often of no small value.

Hirn's experimental work furnishes other valuable data, as may be seen in the accompanying table, abstracted from Ledieu. The method of loss and its distribution are here well exhibited.

HEAT-WASTES.— HIRN.

Kind of engine	Simple	Same	Simple	Same	Comp. Woolf	Same	Comp. Woolf	Same
Jacketed ?	No	Yes	No	No	No	Yes	Yes	Yes
Steam-pressure, atmospheres	5	5	4	4.5	4	4	5	4.5
I. H. P.	67.5	98	129.5	141	106	130	263	690
Ratio of expansion	13.5	9.1	3.9	3.9	4.4	4.4	7.5	4.8
Revolutions per minute	55	55	29	29	23	23	25	75
Proportion of cylinder-condensation	0.33	0.21	0.18	0.12	0.26	0.20	0.19	0.18
Total condensation at entrance	0.49	0.34	0.25	0.11	0.053	0.01	0.091	0.037
Heat restored during expansion	0.18	0.31	0.086	0.018	−0.067	0.075	0.14	0.011
Heat lost during exhaust by re-evaporation	0.29	0.054	0.064	0.085	0.12	0.07	0.016	0.064
Heat lost by jackets	....	0.038	.....	.....	.....	0.098	0.10	0.061
Heat gained by piston-friction	−0.013	0.023	−0.002	0.009	0.005	0.072	0.068	0.089

One of the most satisfactory determinations of the value of the steam-jacket on compounded engines is that of Professor Osborne Reynolds, of Owens College. Manchester, employing the triple-expansion engines of the Whitworth Laboratory.*

The three independent engines combined in the machine were of the following dimensions: †

	CYLINDER.	
	Diam.	Stroke.
No. 1	5 inch.	10 inch.
" 2	8 "	10 "
" 3	12 "	15 "
Air-pump on No. 3	9 "	4½ "
Feed-pump	1½ "	2 "

All were jacketed, sides and heads; steam was carried at 200 lbs. per square inch, and boiler pressure was maintained in all the jackets. The results were the following, with and without the jackets in use:

* *Proceed. Brit. Inst. C. E.*, Dec. 10, 1889.

† *Manual of the Steam Engine*, Vol. I., p. 649.

	With Jackets.	Without.
Coal, per horse-power per hour.....	1.33 to 1.50	1.02 to 1.81
Water, " "	12.68 " 14.10	15.90 " 17.30

The effect of radiation was determined and found somewhat considerable. Deducting this waste, the figures stand:

	With Jackets.	Without.
Coal.....	1.21 to 1.30	1.54 to 1.77
Water.....	11.90 " 12.30	15.10 " 16.60

This is one of the most satisfactory approximations to the ideal engine and to minimum wastes that has ever yet been recorded. In this wonderfully economical little engine, the loss by shutting off the jackets was from 25 to over 35% in fuel consumption; or from 25 to 30% in water expenditure. Of the total heat received, exclusive of radiation, 19.4% was converted into work with jackets in action, and but 15 without; a difference of over 23% of the first quantity, or 29 of the latter. The ideal engine, under similar thermodynamic conditions, would have utilized 23%. The effect of the jacket on the high-pressure cylinder, where the difference of temperature between jacket-steam and initial was small, was found to be slight as affecting cylinder condensation. In No. 2, the effect with a difference of temperature in this respect of 80° Fahr., that condensation was reduced from 30 to 5%, while in No. 3, with a difference of 180° Fahr., such condensation was sensibly zero and the "saturation expansion curve," assumed by Rankine to be attainable by this use of the jacket, was perhaps for the first time produced.

The performance of the engines in this respect is illustrated by the following tables of data obtained by Professor Reynolds.*

In the first three cases, in the first table, the steam-jackets were in use; in the last three, they were disconnected. The difference in result is due to the greater cylinder-condensation in the latter case; the total amounts of which are given in the second table; in which the proportion thus noted ranges from about one-fourth to about one-half.

* *Minutes of Proceedings of the Inst. of C. E.*, 1889; No. 2407.

AREAS OF DIAGRAMS PER POUND OF STEAM AND THERMAL EFFICIENCIES OF ENGINES.

Number of trials	44	33	56	41	35	40
Theoretical area, ft. and lbs.	238,645	233,545	228,420	235,500	233,000	221,860
Measured area, ft. and lbs.	188,096	192,067	192,000	127,545	139,546	144,350
Percentage of theoretical area	79.0	82.0	84.6	54.0	60.0	65.0
Theoretical efficiency, p. c.	23.3	23.2	22.7	23.3	23.2	23.3
Measured efficiency, p. c.	18.5	19.2	19.4	14.1	15.3	15.5
Percentage of theoretical efficiency	79.4	82.6	83.1	60.4	65.9	66.4

CONDENSATION WITHOUT JACKETS.

	Number of the Trial.	Revolutions per Minute.	Ratio of Expansion.	Proportion of total steam condensed at		
				Cut-off.	Mid-stroke.	Release.
Engine No. I.	41	146	2.7	0.40	0.39	0.30
	35	229	2.3	0.29	0.27	0.23
	40	322	2.0	0.22	0.21	0.17
Engine No. II.	41	127	2.4	0.41	0.35	0.29
	35	215	2.4	0.38	0.31	0.26
	40	320	2.2	0.30	0.27	0.14
Engine No. III.	41	109	2.7	0.51	0.48	0.37
	35	184	3.05	0.48	0.47	0.33
	40	276	2.6	0.32	0.36	0.23

The action of the jacket may be considered a method of securing the equivalent of reduced conductivity and heat-storing power in the cylinder-wall. This heat-storage may be computed, where the range of temperature of the surface is known, by the equation of Fourier for the flow of heat into a solid and indefinitely extended mass:

$$Q = a\sqrt{t}(T_1 - T_2);$$

$$= a\sqrt{\frac{1}{N}}\,(T_1 - T_2);$$

in which, for British measures, and time in minutes, $a = 2.45$ for iron.

Could the temperature of the inner wall of the cylinder be held up to T by the action of the jacket, as by employing steam of pressure exceeding that in the engine, $\Delta\, T$ would become

zero, and the result would be precisely the same as making the conductivity of the wall, or its specific heat, or both, zero. It is also seen that the higher the speed the less the quantity of heat stored and wasted by initial condensation. The depth at which the changes of temperature are similar in time to those at the surface, and what may be taken as the limit of sensible penetration of heat, is similarly given by the expression.

$$d = a\sqrt{\frac{1}{N}};$$

in which, for depths in inches and for revolutions per minute, a is approximately 4.5; and the penetration, in this sense of the term, is nearly two inches for each thermal unit of heat abstracted from the steam per unit of area and per degree range of temperature of surface film, at any given speed of rotation. The depths thus attained are, however, usually slight.

Where the cylinder is jacketed, the heat-flow from the jacket is measured by the quantity

$$Q' = a\,t\,(T_1 - T_2) \div d;$$

where the temperatures are those of the two sides of the cylinder-wall and the thickness of the "liner" is d in inches, in the present case, a being, according to Forbes, about 7.5 for time, t, in minutes. But the range of temperature is here one-half that in the preceding expressions; being the difference between the maximum external temperature of the cylinder-wall and the mean internal temperature, and, the symbols being retained and the full range taken, $T_1 - T_2$, the coefficient becomes 3.75. Where t is the time of a half revolution and Q' is the heat transmitted between jacket and cylinder during the period of abstraction of the quantity Q' by the metal, a becomes 1.875 and the quantity flowing in either direction,

$$Q' = 1.875\,(T_1 - T_2) \div d\,N;$$

where, as before, the temperature-range is that of the surfaces of the cylinder-wall.

The following table, computed by Cotterill, following Fourier and Kirsch, gives approximate measures of these quantities.*

* Cotterill, p. 286.

ACTION OF CYLINDER-WALL.

N	25	50	100	200	300	400	600
d	.89	.63	.45	.32	.26	.22	.18
Q	.49	.35	.25	.17	.14	.12	.10
Q'/Q	.16	.11	.08	.06	.05	.04	.03

The variations of temperature-head and of flow due, in all engines, to varying expansion; the different resistances of various metals and surfaces in different engines, and the variations of these quantities in the same engine at different times make all such computations exceedingly uncertain. In the table the figure for Q is reduced to that for *one* degree range of temperature.

Referring to the last equation, we observe that the thinner the cylinder wall, the larger the flow from the jacket, and, presumably, the higher its efficiency. The slower the engine speed, also, the higher this rate of heat flow. The wider the mean range of temperature between extremes, $T_1 - T_2$, the more active the jacket, *i. e.*, the greater the ratio of expansion and the higher the steam-pressure, the more effective the jacket; conclusions already reached by experimental investigation. In the course of the engine cycle the steam, at entrance, gives up to the cylinder the quantity $Q - Q'$ and during the period between the closing of the steam-valve and the next admission of that end of the cylinder, regains the quantity $Q + Q'$. The difference,

$$(Q + Q') - (Q - Q') = 2Q',$$

is the heat received from the jacket in the complete revolution.

These mathematical deductions are rendered still less positive by the fact that the alternate condensation and re-evaporation of films of moisture upon the metal wall, and the action of the non-conducting films of permanently existing liquid—the one phenomenon facilitating, the other obstructing, the transmission of heat between steam and metal—introduce conditions so variable and obscure as to make any conclusions, unless fully confirmed by experimental test, entirely uncertain. Experiment has, however, fully corroborated the general method, thus indicated, as that of heat-transfer in the engine-cylinder. The depths of penetration of the heat-wave between steam and metal and the ratio of quantity of heat thus transmitted to quantity

received from the jacket, Q' / Q, seem, also, to fairly represent the actual rates of flow, as observed in the best representative cases.

M. Dwelshauvers-Dery, analyzing the data supplied by test of the Whitworth Laboratory experimental engine, obtains the following:*

Let Q = heat supplied by the boiler directly;
Q_1 = that supplied by the jacket;
T = total indicated work;
E = rejected heat externally;
$C + c$ = that sent to the condenser.

Then

$$Q + Q_1 = T + E + (C + c);$$
$$\frac{T}{Q + Q_1} + \frac{E_1}{Q + Q_1} + \frac{C + c}{Q + Q_1} = 1.$$

Referring to the six tests, in three of which the jackets were all in use, and in three of which they were on the reservoirs only, and were shut off on the cylinders, the following table is obtained:

HEAT DISTRIBUTION.

Number of trial		44	33	56	41	35	40
Heat of the direct steam,	$\frac{Q}{Q + Q_1}$	0.749	0.789	0.816	0.869	0.893	0.904
Heat of the jacket,	$\frac{Q_1}{Q + Q_1}$	0.251	0.211	0.184	0.131†	0.107†	0.096†
Work,	$\frac{T}{Q + Q_1}$	0.153	0.165	0.171	0.125	0.135	0.136
Radiated waste,	$\frac{E}{Q + Q_1}$	0.127	0.091	0.060	0.040	0.053	0.045
Heat utilized by jacket,	$\frac{Q - E}{Q + Q_1}$	0.124	0.120	0.124	0.091†	0.054†	0.051†
Heat rejected,	$\frac{C + c}{Q + Q_1}$	0.720	0.744	0.769	0.835	0.812	0.819

* *Manual of the Steam Engine*, Vol. I., p. 648.

† Jackets shut off of cylinders only, presumably still doing some work, through slight leakage in cylinders.

Thus, deducting the quantities of heat wasted by external radiation, the jackets supply an almost perfectly uniform quantity of heat, the figures being 12.4%, 12%, and 12.4% in the first three trials; the cause of the greater variation in the last three cases is indeterminate from the available data.

M. Dwelshauvers-Dery finds a gain of 17% produced by the cylinder jackets, by the net gain due their action in promoting efficiency of working, aside from thermo-dynamic efficiency. Studying the famous experiments of M. Hirn at Logelbach, with and without superheating, he finds that superheating produced a gain of 25% by promoting thermo-dynamic efficiency; raising its value from 0.324 to 0.433; while the saturated steam was made more efficient as a working substance by jacketing, and to the extent of 17%, although the purely thermo-dynamic efficiency was actually reduced 1.6%.

The distribution of heat in the course of flow of the two streams of energy entering the working cylinder and the jacket of the engine from the boiler, in the case of the most economical of modern forms of high-pressure multiple-expansion engine, is illustrated by Tables *A*, *B*, *C*, in the Appendix *A*, embodying the results of a thermal analysis of the triple-expansion steam pumping engine of the city of Milwaukee, reported by the writer to the American Society of Mechanical Engineers, as giving, in trials conducted by the Department of Experimental Engineering of Sibley College, Cornell University, under the supervision of Professor Carpenter, a duty of 143,000,000 per 100 lbs. of coal, or 154,000,000 per 1,000 lbs. dry steam, the work performed being 574 I.H.P., 521 D.H.P., and the steam used 11.678 lbs. per I.H.P. per hour, and the heat 217.6 B.T.U. per minute, the steam pressure being 121.6 lbs. above the atmosphere, and the ratio of expansion 19.55.*

The radiation-losses and the quantity of heat rejected from each cylinder, assuming equal distribution of jacket-heat among them, is as follows. The symbols are those customarily used in these thermal analyses, and are explained by the context in the tables.

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* "Maximum Economy of the Contemporary High-pressure Multiple-expansion Steam-Engine:" *Trans. A. S. M. E.*, December, 1893; Vol. XV., No. DLXVI.

RADIATION LOSSES AND HEAT REJECTED PER 100 REVOLUTIONS.

Total heat admitted to high-pressure cylinder	598,800	B. T. U.
Total heat used in all jackets, assuming 1-3 weight of jacket steam to be used in each jacket	40,870	"
Total heat used	639,670	"
Heat rejected from low-pressure cylinder	513,420	"
Total work done	119,652	"
Total radiation loss	6,598	"
	639,670	"
Radiation loss from each cylinder assumed to be equal in all cylinders	2,199	"

Heat rejected from high-pressure cylinder = heat entering + heat supplied by jacket — radiation loss — work done

$$= 598,800 + 13,360 - 2,199 - 36,456$$
$$= 573,505 \text{ B. T. U.}$$

Heat rejected from intermediate-pressure cylinder = heat rejected from high-pressure cylinder + heat supplied by jacket — radiation loss — work done

$$= 572,505 + 13,360 - 2,199 - 35,286$$
$$= 549,380 \text{ B. T. U.}$$

Heat rejected from low-pressure cylinder = heat rejected from intermediate-pressure cylinder + heat supplied by jacket — radiation loss — work done

$$= 549,380 + 14,150 - 2,200 - 47,910$$
$$= 513,420 \text{ B. T. U.}$$
$$= K + K' \text{ (see low-pressure analysis in tables).}$$

The appended tables give the computed distribution of heat in the working charge of steam, and exhibits the interchange of heat between that charge and the jacket-steam, the quantities being computed as a matter of convenience, as is usual, for 100 revolutions. The real distribution of the radiation losses among the cylinders, and of the heat received *via* the jacket, is indeterminable, and it is assumed that both quantities are equally divided among the three cylinders. The probabilities are that the jacket-supply comes in much less proportion from the high-pressure jacket than from either the intermediate or the low-pressure; on the latter, the jacket-steam is reduced to a comparatively low pressure by means of a reducing-valve in the passage of supply. It is seen, above, that the heat-supply consists of 598.8 B.T.U. received from the boiler with the working charge, per revolution, and 408 by the jackets; which is distributed, 66 B.T.U. as radiation loss, 342 B.T.U. as jacket-waste (*i. e.*, lost in the partial suppression of the greater waste by initial condensation), and 1,197 B.T.U. converted into work. The

sum 6,397 B.T.U. received per revolution, divided by the work per revolution, the equivalent of 1,197 B.T.U., is a measure of the cost of the work performed, and its reciprocal is the efficiency of the system. The jacket-heat constitutes, as seen, 9.22% of the total from the boiler; and this reduces the initial condensation from what would probably have been above 30% to 12, returning its own value plus above 12%, or about 20%, through reduction of initial wastes and addition to the work of expansion; the net gain being effected by having the otherwise anticipated cylinder condensation. In the tables, the losses are marked positive when heat passes from the working charge to the cylinder-wall, negative when received into the steam within the cylinder. It is seen, in the first high-pressure table, for example, that the charge yields 62,629 B. T. U. to the metal during admission, regains 46,990 during expansion, and entirely wastes 27,862 units during the period of exhaust, and 1,064 during compression. The initial condensation is 98.95 − 86.77 = 12.18%, the charge losing 0.46% of its heat to the cylinder-wall in admission, regaining 7.86 and 4.65% during expansion and exhaust. Six per cent. of the total is utilized in work, 0.37% is lost externally, and this one cylinder has 0.68% of the efficiency of the ideal machine of similar temperature-range.*

In the three cylinders, successively, the initial condensation is seen to be, respectively, 12.18%, 13.55%, and 17.80%, exhibiting in unusual degree the correct apportionment of this waste among the cylinders. The more equal this loss, the more perfectly does the engine approximate the ideal action of the multiple-cylinder engine;† the heat supplied, to be thus wasted from the one cylinder, being rejected from it in precisely the quantity needed to meet the demand for waste heat in this phase of the cycle in the next cylinder. This special feature of operation of the engine, the efficiency of its jackets, its effective covering with non-conductors, its tight condenser, and its small clearances (1.4%, 1.5% and 0.77%), are the secrets of its previ-

* The large plate accompanying the paper to which reference is here made, and which exhibits the indicator diagrams *in fac-simile*, and combined with the appropriate adiabatic and saturation curves, illustrates graphically, in a very beautiful manner, the changing values of these quantities and the varying quality of the steam passing through the engine.

† "Philosophy of the Multiple-Cylinder Engine:" R. H. Thurston. *Transactions American Society Mechanical Engineers*, Vol. XI., No. CCCLXII., 1890.

ously unapproached economy. The total efficiency of the engine is 0.68% that of the corresponding Carnot cycle, and 0.75% that of the ideal Rankine cycle of similar steam and heat limits. The figures here given may be taken as representative of the best current practice, and of the extent to which jacket action is effective under such conditions of maximum efficiency.

Similarly Appendix B presents the results of thermal analysis of the experimental engine of the Massachusetts Institute of Technology, which lead to the deductions:

"An investigation of the results in the table shows that the difference in economy depends on the cut-off of the high-pressure cylinder, which is at about $\frac{1}{5}$ stroke for the first two, and at about $\frac{1}{4}$ stroke for the last two. The quantities that are most notably affected by the change of cut-off are the heat furnished by the high-pressure jacket and the condition of the steam, and interchanges of heat in the high-pressure cylinder. Thus the heat supplied by the high-pressure jacket for the tests, with the cut-off at $\frac{1}{4}$ stroke, is about half that furnished in the tests at $\frac{1}{5}$ stroke; which finds explanation or confirmation in the facts, that the steam is much dryer at cut-off with the longer cut-off, and that both the heat absorbed and yielded by the walls of the cylinder is notably less with that cut-off."

* * * * * * * *

"It is notable that the steam becomes dryer in its course through the engine, under the influence of the thorough steam-jacketing, with steam at boiler pressure, and that it is practically dry at release, in both the intermediate and low-pressure cylinders. In some of the tests it appears to be superheated at this point, and in one, even at cut-off in the low-pressure cylinder. The superheating is in no case large, and as it is accompanied by a small positive value for Q_c, the exhaust-waste, it may be that the steam was really dry and saturated at those points, though superheating by jackets filled with steam at so high a pressure is not impossible. Since a positive value for Q_c indicates that the walls absorbed heat during exhaust, such a result, if large, would be absurd; but in these tests it may properly be looked upon as the error, being in no case larger than about 2% of the value of Q; if the steam at release is either dry or superheated, the value of Q must be very small, or zero.

"It is most remarkable that the heat furnished by the jackets is nearly as large in amount as that changed into work in the

tests with the cut-off at $\frac{1}{4}$ stroke, and is actually larger for the tests with the cut-off at $\frac{1}{8}$ stroke. Also, that the heat lost by radiation is one-third as much as the heat changed into work, if not more; the engine, being arranged for convenient experimental work, probably has a large radiation, though it is thoroughly wrapped and lagged on all hot surfaces.*

METHODS OF INVESTIGATION should be standard and exact. A committee of the Institute of Mechanical Engineers has suggested the following, as instructions for experiments to determine effectiveness of steam-jackets: †

(1) The date, place, and duration of each experiment should be recorded, and the names of the experimenters.

(2) In the description of the engine it should be stated whether it is single cylinder, compound, or triple expansion, horizontal or vertical, condensing or non-condensing, and the diameters and strokes of the cylinders should be given, and the kind of valves mentioned.

The following particulars should also be furnished: Manner in which each cylinder and receiver is jacketed, giving dimensions for enabling the jacketed and unjacketed areas of the internal surfaces of each to be calculated. Particulars as to how the steam is supplied to the jackets, and how they are drained, giving the lengths and dimensions of the supply pipes. Details of how the outer walls of the jackets are protected.

(3) A sufficient number of measuring tanks, etc., should be provided to allow the feed-water and the discharges from the jackets, drain-pipes, etc., to be measured separately.

(4) Each body-jacket should be provided with a pressure-gauge, and also with two cocks, one at the highest point and the other at the lowest, to test for air, steam, or water during the experiments. Each of the other jackets should also be provided with an air-cock at its highest point, so that if air is present it may be allowed to escape from time to time.

(5) A separate indicator should be used for each end of each cylinder, and the engine should be fitted with a speed-counter.

(6) Before the experiments commence, the valves and pistons should be tested for leakage by blocking the engines in different positions and opening the indicator cocks.

(7) The body-jackets should also be tested for leakage at the

* *Transactions American Society Mechanical Engineers*, 1891.

† *Proceedings Institute Mechanical Engineers*, 1892.

usual steam-pressures, the pistons and cylinder-covers being removed if possible, or by steam being admitted to the jackets and the indicator cocks opened before steam is admitted to the cylinder.

The quantity of steam condensed in the jackets when the engine is standing should be ascertained by admitting steam to them at the usual pressures, and measuring the discharge from each.

(8) Throughout the whole of the experiments the quantities of feed-water, circulating or injection water, jacket water, drain water leakage from valves or stuffing-boxes, and water from the steam separator should be measured. The discharge from each jacket should be taken separately, and the temperature of the feed water and of the circulating or injection water should be noted.

(9) The readings of the speed-counter and the pressure-gauges should be all noted at regular intervals. Sets of indicator diagrams should be taken at regular intervals, each set consisting of a diagram from each end of each cylinder, taken as far as practicable simultaneously. The highest and lowest pressure in each cylinder should be ascertained from the diagrams and recorded, for allowing the range of temperature in each cylinder to be calculated. The mean effective pressures and the ratio of expansion should be calculated from the diagrams and recorded, and the indicated horse-powers worked out. The percentage of moisture in the steam should be recorded, if known. The kind of lubricant used for the cylinders should be stated, and the same quantity per hour should, as far as possible, be supplied during comparative trials with and without steam in the jackets.

COMPUTATIONS OF PROBABLE EFFICIENCY OF JACKETED ENGINES may be made to give very fairly approximate results, good representative practice being assumed.

The following table * represents the results of computations of probable efficiency and performance, on the assumption that the initial pressure is 250 lbs. per square inch, absolute, the internal wastes as found in experimental work already referred to and measured by the expression $1 + 0.075 \sqrt{r}$, the engine being a jacketed, tandem-compound engine; these wastes correspond with those of a single jacketed cylinder of moderate

* *Transactions American Society Mechanical Engineers*, CCCCLI., Vol. XII., 1891. *Manual*, Vol. I., § 155, pp. 641-646.

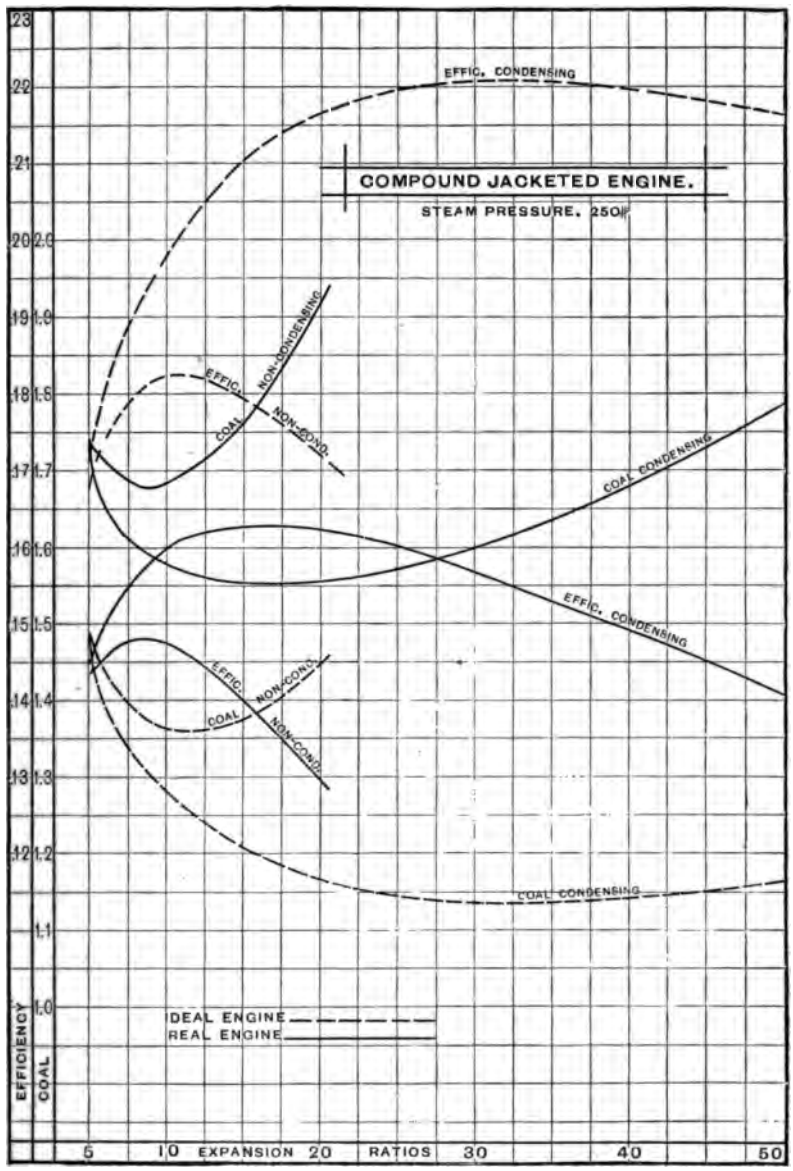

FIG. 247.—Engine Efficiencies.

size and under usual conditions of operation.* Back-pressures are taken as 5 lbs. condensing, and 16 non-condensing, feed-tem-

* M. Haton de la Goupillière coincides with Sinigaglia; who says that this function was first proposed by the author, and subsequently confirmed by direct experiment at Sandy Hook and elsewhere.—*Cours des Machines*, Vol. II.

COMPOUND STEAM-ENGINE—JACKETED.

NON-CONDENSING.

IDEAL CASE.																				REAL CASE.			
r	v_1	$v_2 = rv_1$	p_2	p_3	$v_2(p_2-p_3)$	U_1	U_2	$U_1 - U_2$	$U' = v_1 - v_2 + v_2(p_2-v_3)$	H_2	(Feed = 203) h_4	$h = U_1 - U_2 + H_2 - h_4$	$H_1 - h_4$	Heat supplied by Jacket per lb. steam. Q $h - (H_1 - h_4)$	Efficiency $= \frac{U'}{h}$	Lbs. H_2 Q worked in cyl. pr. I. H. P. pr. hr. S $\frac{1980000}{U'}$	M Coal to evap. S $\frac{S}{10 \text{ \& } 9}$	N Coal to supply heat supplied Jacket. $\frac{QM}{H_1 - h_4}$	Total fuel. $M+N$	$1 + 0.075\sqrt{r}$	Efficiency.	Water per I. H. P. pr. hr.	Coal per I. H. P. pr. hr.
5	1.84	9.2	6494	2304	38548	420280	312633	107647	146195	899654	132360	875141	797251	77890	.1671	13.54	1.354	.1354	1.489	1.168	.1430	15.815	1.739
8	"	14.72	3931	"	23949	"	283189	137091	161040	892916	"	897647	"	100396	.1794	12.29	1.229	.1598	1.389	1.212	.1480	14.895	1.683
10	"	18.40	3102	"	14683	"	269523	150757	165440	889960	"	908357	"	111106	.1821	11.96	1.196	.1674	1.363	1.237	.1472	14.795	1.686
15	"	27.60	1987	"	— 8749	"	245228	175052	166303	884604	"	927296	"	130045	.1793	11.91	1.191	.1906	1.382	1.290	.1392	15.364	1.783
20	"	36.80	1483	"	—30213	"	228319	191961	161718	881093	"	940694	"	143443	.1720	12.24	1.224	.2208	1.444	1.335	.1290	16.340	1.928
CONDENSING.																							
5	18.4	9.20	64.94	720	53121	420280	312633	107647	160768	899654	55612	951889	873999	77890	.1689	12.315	1.368	.122	1.490	1.168	.145	14.384	1.740
10	"	18.40	3102	"	43829	"	269523	150757	194586	889960	"	985105	"	111106	.1975	10.170	1.130	.142	1.272	1.237	.1597	12.580	1.573
20	"	36.8	1483	"	28078	"	228319	191961	220039	881093	"	1017442	"	143443	.2163	8.995	.999	172	1.171	1.335	.1620	12.008	1.561
30	"	55.2	961.	"	13303	"	204958	215322	228825	876434	"	1080144	"	162145	.2207	8.660	.962	.1732	1.135	1.411	.1564	12.360	1.601
40	"	73.6	709.9	"	— 743	"	189023	231257	230514	873373	"	1049018	"	175019	.2197	8.590	.954	.189	1.143	1.474	.1491	12.662	1.685
50	"	92.0	559.9	"	—14729	"	176665	243625	228896	871067	"	1059080	"	185081	.2161	8.650	.961	.208	1.169	1.520	.1410	13.235	1.789

p_1 = 36,000 lbs. per sq. ft. = 250 lbs. per sq. in.

peratures as 104° Fahr. and 203° Fahr., respectively, and the evaporations as approximately 9 and 10. The value of the co-effi-

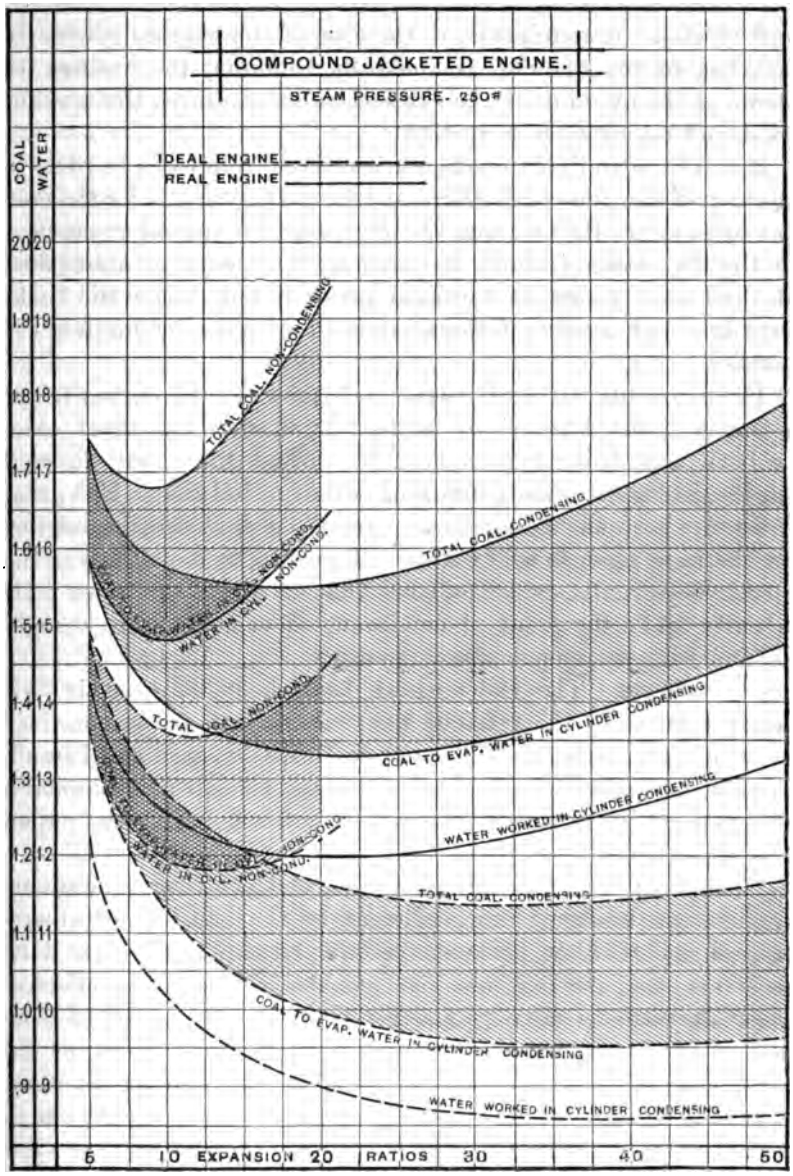

FIG. 248.—Fuel and Water Consumption.

cient, 0.075 $\sqrt{r}$, is that customarily taken where it may be assumed that "compounding" and jacketing have reduced the

internal wastes to about one-half the usual minimum with unjacketed simple engines of this size and class.

It is obvious that, in the computation of probable efficiency, and of steam consumption, in the case of the engines efficiently jacketed in the manner assumed by Rankine, the volume of steam at the opening of the exhaust-valve measures the amount used, and requires no correction.

It will be seen that the efficiencies range from 16.7% to 18.2% in the case of the non-condensing, and from 16.9% to 22% for the condensing engine, the maximum being found at a ratio of expansion, in the first case, of about 10%, and in the second of about 30%. Beyond these ratios the terminal pressure falls below the back-pressure, and a waste follows, instead of gain, by further expansion.

These results are still better exhibited by the curves (Figs. 5 and 6) plotted from the numerical values; the ideal case, in both sets, being represented by dotted lines, and the real engine giving the widely different curves in full line. The great difference between the condensing and the non-condensing engine, for the ideal case, is well shown. The gain by expansion in the former case continues far beyond that at which the latter finds a limit; while the point of maximum effect is far more sharply defined with the non-condensing engine.

CONCLUSIONS.—The views which have been deduced in this paper relative to the value of the steam-jacket are summarized in the recently issued pocket-book of Messrs. Seaton and Roundthwaite as concisely and clearly as possible. In this summary, received by the writer at the moment of concluding the paper, it is remarked: *

"The most recent investigations show that all types of steam-engines are rendered more efficient by the addition of steam-jackets, and that the more completely the surfaces of cylinders, receivers, etc., are jacketed, the greater is the saving effected. This amounts to saying that, for every pound of steam condensed in the jackets, some greater quantity is saved in the cylinders. The ratio of steam saved in the cylinder to steam expended in the jackets varies from a little under two to one in some types of engines, to over five to one in other types. The gain that may be expected to result from jacketing an engine

* Page 12.

depends upon such a multitude of considerations, relating not only to the design of the engine and of the boiler, but also to the management of the machinery under steam, that it can only be very generally stated as lying between five and twenty-five per cent. of the total feed-water evaporated; but in the case of modern marine machinery of good construction it is not probable that the gain is over ten per cent. of the total feed-water evaporated."

From what has preceded it is sufficiently obvious that when jackets are used care should be taken to meet the following essential conditions of efficient and economical working: †

(1) The jacket should be provided with ample supply-pipes and with effective traps or other drainage arrangements, and for removal of air as well as water. If the jacket can be made to drain back to the boiler, that plan should always be adopted.

(2) They should be kept supplied with steam at a pressure equal to that in the boiler.

(3) All surfaces exposed to full-pressure steam should be jacketed, if practicable.

(4) The jacket itself should be very carefully and thoroughly lagged, and so made secure against serious external waste of heat.

(5) Provision for safe expansions and contractions should be very carefully made.

(6) It should be seen that the jacket steam has everywhere complete contact with the inner or working cylinder, and that all water precipitated therefrom may promptly and completely drain away.

(7) The walls of the cylinder, or "liner," should be as thin as practicable, and yet safe; all core-spaces should be free and clear; all core-sand thoroughly removed; no pockets should exist in which water may gather, and all fits and joints should be made with extreme care.

(8) It is probably wise to jacket all the cylinders of a multiple-cylinder engine if maximum economy is sought.

(9) The jacket, in cases in which steam passes through it on the way to the working cylinder should be designed and proportioned to act as an effective separator. It may then give good results by the currents of steam sweeping the cylinder surface free from films of gathering water.

† *Manual*: Section 166, p. 668, Vol. I.

A jacket through which the steam entering the cylinder should pass would have a great advantage in efficiency of heat-transfer; but unless the entrained water and condensed steam could be completely removed, it would cause counterbalancing, and probably greater, losses, as compared with the usual arrangement, by carrying that water into the engine to exaggerate wastes.

"In all cases, and under all conditions, the use of a steam-jacket is 'a violation of the fundamental law of maximum efficiency of heat-engines, which requires that they should receive all their heat at the maximum and give it out at the minimum temperature, and not, as in the case of an engine with a steam-jacket, at temperatures between these, and at times when the heat imparted lessens efficiency, which it evidently must do at and near the end of the stroke.' It is a necessary evil, justified only by the conditions affecting the use and the construction of the engine. The advantage to be derived thus varies according to circumstances, and the jacket may not only sometimes be useless, but wasteful. The necessity for a careful study of the conditions of use, of care in its application, and of exact determination of its value, is evident." *

* *Ibidem*, § 166, pp. 670–671.

APPENDIX A.

MILWAUKEE PUMPING ENGINE.

A.

THERMAL ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS. HIGH-PRESSURE CYLINDER (JACKET STEAM EXCLUDED) WITH FIRST RECEIVER.

Quantities.	Symbols.	Formulæ.	
1. Steam from boiler entering working cylinder, pounds	M	.....	504.5
2. Steam at admission, pounds	M_c	$100\,(V_c + C_c X_c)$	18.158
3. Steam at admission, pounds	M_0	$100\,(V_c + V_0 + v_0$	.....
4. Steam used by calorimeter, pounds	.....	.....	.....
5. Steam, total, pounds	$M + M_0$	.....	522.658
6. Heat of condensed steam	K'	Mq_g	.....
7. Condensed water, pounds	G	.....	.....
8. Heat given to condensing water	K	$G\,(q_k - q_l)$ (Heat rejected.)	573,505
9. Heat supplied to engine	Q	$M\,(xr + q)$	598,800
10. Sensible heat at admission	H_0	$M_0 q_0$	5,819
11. Internal heat at admission	H_0'	$100\,\frac{V_c + V_0}{v_0}\,\rho_0$	14,270
12. Sensible heat at cut-off	H_1	$(M + M_0)\,q_1$	167,400
13. Internal heat at cut-off	H_1'	$100\,\frac{V_c + V_2}{v_1}\,\rho_1$	356,900
14. Sensible heat at release	H_2	$(M + M_0)\,q_2$	128,700
15. Internal heat at release	H_2'	$100\,\frac{V_c + V_2}{v_2}\,\rho_2$	411,300
16. Sensible heat, beginning of compression	H_3	$M_0 q_3$	4,527
17. Internal heat, beginning of compression	H_3'	$100\,\frac{V_c + V_3}{v_3}\,\rho_3$	15,700
18. Cylinder loss during admission	Q_a	$Q + H_0 + H_0' - H_1 - H_1' - A\,W_a$	+62,629
19. Cylinder loss during expansion	Q_b	$H_1 + H_1' - H_2 - H_2' - A\,W_b$	−46,990
20. Cylinder loss during exhaust	Q_c	$H_2 + H_2' - H_3 - H_3' - K - K' - A\,W_c$	−27,862
21. Cylinder loss during compression	Q_d	$H_3 + H_3' - H_0 - H_0' - A\,W_d$	−1,064
22. Heat discharged, and work	B	$K + K' + A\,W$	609,061
23. Jacket*	D	$Q - B$	−11,161
24. Jacket*	D'	$Q_a + Q_b + Q_c + Q_d$	−11,161
25. Quality of steam entering	x	per calorimeter ..... per cent	98.95
26. Quality of steam at cut-off	x_1	$100\,\frac{V_c V_1}{(M + M_0)\,v_1}$ ..... "	86.77
27. Quality of steam at release	x_2	$100\,\frac{V_c + V_2}{(M + M_0)\,v_2}$ ..... "	93.3
28. Quality of steam at compression	.....	$100\,\frac{V_c + V_3}{M_0 v_3}$ ..... "	.....
29. Quality of steam at admission	x_0	per calorimeter ..... "	.....
30. Quality of steam at exhaust	x_5	$\left(\frac{K + K'}{M - Mx} - q_5\right) \div r_5$ ..... "	96.8
31. Heat lost, admission	a	$Q_a \div Q$ ..... "	10.46
32. Heat restored, expansion	b	$Q_b \div Q$ ..... "	−7.86
33. Heat rejected, exhaust	c	$Q_c \div Q$ ..... "	−4.65
34. Heat lost, compression	d	$Q_d \div Q$ ..... "	.18
35. Heat utilized, work	w	$\frac{W}{778} \div Q$ ..... "	6.067
36. Heat lost, radiation	R	radiation $\div Q$ ..... "	0 367
37. Ratio, radiation to work	.....	$R \div w$	.0603
38. Ratio, cylinder condensation to work	.....	$a \div w$	1.72
39. Thermo-dynamic efficiency	E	$(t - t_3) \div (460 + t)$ ..... per cent.	8.96
40. Actual efficiency	E_1	$A\,W \div Q$ ..... "	6.087
41. Efficiency compared with ideal	E'	$E_1 \div E$ ..... "	68.0

Special symbols Vc = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A = \frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* This quantity is the difference between the heat lost by radiation and that received from the jacket. The negative sign shows that more heat is received than lost.

B.

THERMAL ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS—INTERMEDIATE CYLINDER (JACKET STEAM EXCLUDED) WITH SECOND RECEIVER.

Quantities.	Symbols.	Formulæ.		
1. Steam entering cylinder, pounds	M	.....		504.5
2. Steam at admission, pounds	M_c	$100\,(V_c+V_0)\div v_0$		22.33
3. Steam used by calorimeter, pounds	.....	.....		.....
4. Steam, total, pounds	$M+M_0$	.....		526.83
5. Heat of condensed steam	K'	Mq_g		.....
6. Heat rejected	G	.....		549,380
7. Heat given to condensing water	K	$G\,(q_k-q_i)$		.....
8. Heat supplied to engine	Q	$M\,(xr+q)$		573,505
9. Sensible heat at admission	H_0	M_0q_0		5,534
10. Internal heat at admission	H_0'	$100\,\frac{V_c+V_0}{v_0}\,\rho_0$		18,780
11. Sensible heat at cut-off	H_1	$(M+M_0)\,q_1$		125,600
12. Internal heat at cut-off	H_1'	$100\,\frac{V_c+V_2}{v_1}\,\rho_1$		381,400
13. Sensible heat at release	H_2	$(M+M_0)\,q_2$		95,400
14. Internal heat at release	H_2'	$100\,\frac{V_c+V_2}{v_2}\,\rho_2$		434,100
15. Sensible heat, beginning of compression	H_3	M_0q_3		4,201
16. Internal heat, beginning of compression	H_3'	$100\,\frac{V_c+V_3}{v_3}\,\rho_3$		18,420
17. Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-A\,W_a$		67,439
18. Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-A\,W_b$		−35,755
19. Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-A\,W_c$		−41,465
20. Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-A\,W_d$		−1,380
21. Heat discharged, and work	B	$K+K'+A\,W$		584,666
22. Jacket*	D	$Q-B$		−11,161
23. Jacket*	D'	$Q_a+Q_b+Q_c+Q_d$		−11,161
24. Quality of steam entering	x	per calorimeter	per cent.	.....
25. Quality of steam at cut-off	x_2	$100\,\frac{V_cV_1}{(M+M_0)\,v_1}$	"	85.4
26. Quality of steam at release	x_1	$100\,\frac{V_c+V_2}{(M+M_0)\,v_2}$	"	93.2
27. Quality of steam at compression	.....	$100\,\frac{V_c+V_3}{M_0r_3}$	"	.....
28. Quality of steam at admission	x_0	per calorimeter	"	.....
29. Quality of steam in exhaust	x_5	$\left(\frac{K+K'}{M-M_x}-q_5\right)\div r_5$	"	94.0
30. Heat lost, admission	a	$Q_a\div Q$	"	11.75
31. Heat restored, expansion	b	$Q_b\div Q$	"	−6.28
32. Heat rejected, exhaust	c	$Q_c\div Q$	"	−7.23
33. Heat lost, compression	d	$Q_d\div Q$	"	−0.24
34. Heat utilized, work	w	$\frac{W}{778}\div Q$	"	6.155
35. Heat lost, radiation	R	Radiation $\div\,Q$	"	.384
36. Ratio, radiation to work	.....	$R\div w$		.0623
37. Ratio, cylinder condensation to work	.....	$a\div w$		1.795
38. Thermo-dynamic efficiency	E	$(t-t_5)\div(460+t)$	per cent.	8.99
39. Actual efficiency	E_1	$A\,W\div Q$	"	6.155
40. Efficiency compared with ideal	E'	$E_1\div E$	"	68.55

Special symbols, V_c = volume clearance, t = measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A=\frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* Same explanation as for high-pressure cylinder. Radiation assumed equal for all three cylinders, and heat received from jackets assumed equal in high and intermediate pressure cylinders.

C.

THERMAL ANALYSIS—DATA AND RESULTS PER 100 REVOLUTIONS—LOW-PRESSURE CYLINDER (JACKET STEAM EXCLUDED).

Quantities.	Symbols.	Formulæ.		
1. Steam entering working cylinder, pounds	M	....		504.5
2. Steam at admission, pounds	M_0	$100(V_c+V_0)\div v_0$		9.61
3. Steam used by calorimeter, pounds	...	...		...
4. Steam, total, pounds	$M+M_0$	...		514.11
5. Heat of condensed steam	K'	Mq_g		38,920
6. Condensed water, pounds	G	...		8,184
7. Heat given to condensing water	K	$G(q_k-q_i)$		474,500
8. Heat supplied to engine	Q	$M(xr+q)$		549,380
9. Sensible heat at admission	H_0	M_0q_0		1,746
10. Internal heat at admission	H_0'	$100\frac{V_c+V_0}{v_0}\rho_0$		8,590
11. Sensible heat at cut-off	H_1	$(M+M_0)q_1$		93,450
12. Internal heat at cut-off	H_1'	$100\frac{V_c+V_1}{v_1}\rho_1$		353,600
13. Sensible heat at release	H_2	$(M+M_0)q_2$		68,300
14. Internal heat at release	H_2'	$100\frac{V_c+V_2}{v_2}\rho_2$		399,800
15. Sensible heat, beginning of compression	H_3	M_0q_3		324.2
16. Internal heat, beginning of compression	H_3'	$100\frac{V_c+V_3}{v_3}\rho_3$		3,594
17. Cylinder loss during admission	Q_a	$Q+H_0+H_0'-H_1-H_1'-AW_a$		11,307
18. Cylinder loss during expansion	Q_b	$H_1+H_1'-H_2-H_2'-AW_b$		464
19. Cylinder loss during exhaust	Q_c	$H_2+H_2'-H_3-H_3'-K-K'-AW_c$		−116,818.2
20. Cylinder loss during compression	Q_d	$H_3+H_3'-H_0-H_0'-AW_d$		−6,902.8
21. Heat discharged, and work	B	$K+K'+AW$		561,330
22. Jacket*	D	$Q-B$		−11,950
23. Jacket*	D'	$Q_a+Q_b+Q_c+Q_d$		−11,950
24. Quality of steam entering	x	per calorimeter	per cent.	...
25. Quality of steam at cut-off	x_2	$100\frac{V_cV_1}{(M+M_0)v_1}$	"	81.15
26. Quality of steam at release	x_2	$100\frac{V_c+V_2}{(M+M_0)v_2}$	"	83.33
27. Quality of steam at compression	...	$100\frac{V_c+V_3}{M_0v_3}$	"	...
28. Quality of steam at admission	x_0	per calorimeter	"	...
29. Quality of steam in exhaust	x_5	$\left(\frac{K+K'}{M-M_x}-q_5\right)\div r_5$	"	89.95
30. Heat lost, admission	a	$Q_a\div Q$	"	2.056
31. Heat restored, expansion	b	$Q_b\div Q$	"	.084
32. Heat rejected, exhaust	c	$Q_c\div Q$	"	−21.22
33. Heat lost, compression	d	$Q_d\div Q$	"	−1.256
34. Heat utilized, work	w	$\frac{W}{778}\div Q$	"	8.7
35. Heat lost, radiation	R	Radiation $\div Q$	"	.40
36. Ratio, radiation to work	...	$R\div w$		.046
37. Ratio, cylinder condensation to work	...	$a\div w$		.236
38. Thermo-dynamic efficiency	E	$(t-t_5)\div(460+t)$	per cent.	10.98
39. Actual efficiency	E_1	$AW\div Q$	"	8.70
40. Efficiency compared with ideal	E'	$E_1\div E$	"	79.18

Special symbols, V_c = volume clearance, $= t$ measured temperature. Subscript 5 applies to exhaust, i to injection, k to discharge, g to air-pump discharge. $A = \frac{1}{778}$.

Correct for steam used by calorimeter, when necessary.

* Same explanation as for high-pressure cylinder.

APPENDIX B.

THERMAL ANALYSIS OF TRIPLE-EXPANSION ENGINE.

C. H. PEABODY.

	I.	II.	III.	IV.
Duration of test, minutes	60	60	60	60
Total number of revolutions	5299	5228	5173	5148
Revolutions per minute	88.3	87.1	86.2	85.8
Steam consumption during test, pounds:				
Passing through cylinders	1193	1157	1234	1305
Condensation in h. p. jacket	57	50	29	30
" first receiver jacket	61	64	69	72
Condensation in inter. jacket	85	93	97	105
" second receiver jacket	53	50	52	51
Condensation in l. p. jacket	89	76	90	87
Total	1538	1489	1571	1650
Condensing water for test, pounds	22847	22186	20244	20252
Priming, by calorimeter	0.018	0.012	0.011	0.012
Temperatures, Fahrenheit:				
Condensed steam	95.4	92.1	102.4	105.3
Condensing water, cold	41.9	42.1	43.0	42.8
" " hot	96.1	96.6	106.3	109.6
Pressure of the atmosphere, by the barometer, pounds per sq. inch	14.8	14.8	14.7	14.7
Boiler pressure, pounds per square inch, absolute	155.3	155.5	156.9	157.7
Vacuum in condenser, inches of mercury	25.0	25.1	24.1	23.9
Events of the stroke:				
High-pressure cylinder—				
Cut-off, crank end	0.192	0.194	0.245	0.283
" head end	0.215	0.205	0.271	0.305
Release, both ends	1.00	1.00	1.00	1.00
Compression, crank end	0.05	0.05	0.04	0.04
" head end	0.05	0.05	0.05	0.05
Intermediate cylinder—				
Cut-off, both ends	0.29	0.29	0.29	0.29
Release, both ends	1.00	1.00	1.00	1.00
Compression, crank end	0.03	0.03	0.03	0.03
" head end	0.04	0.04	0.04	0.04
Low-pressure cylinder—				
Cut-off, crank end	0.38	0.38	0.38	0.38
" head end	0.39	0.39	0.39	0.39
Release, both ends	1.00	1.00	1.00	1.00

	I.	II.	III.	IV.
Absolute pressures in the cylinder, pounds per square inch:				
High-pressure cylinder—				
Cut-off, crank end	145.9	145.9	138.8	138.3
" head end	143.2	143.1	140.3	140.6
Release, crank end	41.3	41.5	44.7	48.4
" head end	41.5	40.5	45.7	49.8
Compression, crank end	43.7	45.3	48.5	53.2
" head end	48.7	47.9	54.5	62.0
Admission, crank end	64.5	68.8	72.2	81.2
" head end	75.3	74.8	86.7	97.8
Intermediate cylinder—				
Cut-off, crank end	37.2	37.6	38.6	40.9
" head end	35.0	35.3	39.6	42.6
Release, crank end	13.6	14.2	14.7	16.0
" head end	13.4	13.8	14.9	16.0
Compression, crank end	16.3	17.3	18.2	19.0
" head end	17.9	18.8	20.3	22.4
Admission, crank end	20.4	20.8	22.2	23.1
" head end	21.1	22.8	24.2	26.7
Low-pressure cylinder—				
Cut-off, crank end	12.1	12.6	12.4	13.2
" head end	12.0	12.4	13.1	14.0
Release, crank end	5.6	5.3	5.1	5.7
" head end	5.4	5.8	5.9	6.4
Compression and admission—				
crank end	3.7	3.8	4.1	4.2
head end	4.3	4.5	4.6	4.7
Heat equivalents of external work, B. T. U., from areas on indicator diagram to line of absolute vacuum:				
High-pressure cylinder—				
During admission, $A\ W_a$—				
crank end	5.71	5.78	7.00	8.19
head end	6.61	6.37	8.42	9.50
During expansion, $A\ W_b$—				
crank end	10.65	10.76	10.40	10.25
head end	10.81	11.04	11.22	11.09
During exhaust, $A\ W_c$—				
crank end	7.73	7.89	8.44	9.02
head end	8.08	8.15	9.04	9.66
During compression, $A\ W_d$—				
crank end	0.48	0.60	0.49	0.50
head end	0.62	0.64	0.73	0.81
Intermediate cylinder—				
During admission, $A\ W_a$—				
crank end	7.58	7.57	7.98	8.64
head end	7.43	7.55	8.46	9.10
During expansion, $A\ W_b$—				
crank end	9.54	9.54	9.91	10.64
head end	9.22	9.31	10.37	11.44

	I.	II.	III.	IV.
Intermediate cylinder—				
During exhaust, $A\ W_c$—				
crank end.	9.27	9.47	9.64	10.54
head end..	9.27	9.47	10.18	10.84
During compression, $A\ W_d$—				
crank end.	0.39	0.43	0.57	0.46
head end..	0.60	0.70	0.78	0.84
Low-pressure cylinder—				
During admission, $A\ W_a$—				
crank end.	7.75	7.95	8.33	8.97
head end..	7.99	8.19	8.66	9.39
During expansion, $A\ W_b$—				
crank end.	6.83	7.10	6.86	7.45
head end..	6.87	7.12	7.34	7.87
During exhaust, $A\ W_c$—				
crank end.	5.08	5.08	4.62	5.09
head end..	5.08	5.16	4.81	5.00
During compression, $A\ W_d$—				
crank end.	0.00	0.00	0.00	0.00
head end..	0.00	0.00	0.00	0.00
Quality of the steam in the cylinder. At admission and at compression the steam was assumed to be dry and saturated:				
High-pressure cylinder—				
At cut-off ..... x_1	0.785	0.784	0.848	0.875
At release ..... x_2	0.899	0.903	0.920	0.931
Intermediate cylinder—				
At cut-off ..... x_1	0.899	0.912	0.906	0.908
At release ..... x_2	0.994	super-heated.	super-heated.	super-heated.
Low-pressure cylinder—				
At cut-off ..... x_1	0.978	super-heated.	0.970	0.974
At release ..... x_2	super-heated.	super-heated.	super-heated.	super-heated.
Interchanges of heat between the steam and the walls of the cylinders, in B. T. U. Quantities affected by the positive sign are absorbed by the cylinder walls; quantities affected by the negative sign are yielded by the walls:				
High-pressure cylinder—				
Brought in by steam ... Q	132.93	130.77	141.11	149.84
During admission ..... Q_a	23.54	23.43	17.49	14.93
During expansion ..... Q_b	−18.69	−19.28	−15.33	−14.03
During exhaust ..... Q_c	−8.36	−7.22	−3.50	−2.38
During compression .... Q_d	0.45	0.51	0.49	0.52
Supplied by jacket ..... Q_j	4.56	4.08	2.39	2.50
Lost by radiation ..... Q_e	1.50	1.52	1.54	1.54
First intermediate receiver—				
Supplied by jacket ..... Q_{jR} ...	4.92	5.20	5.67	5.95
Lost by radiation ..... Q_{eR} ...	0.58	0.58	0.59	0.59

	I.	II.	III.	IV.
Intermediate cylinder—				
Brought in by steam... Q'.....	131.89	129.61	137.87	146.64
During admission..... Q'_a....	13.62	11.74	11.33	11.75
During expansion..... Q'_b....	−18.65	−18.84	−20.30	−21.88
During exhaust..... Q'_c....	0.22	1.57	2.88	3.41
During compression.... Q'_d....	0.44	0.51	0.62	0.59
Supplied by jacket..... Q'_j....	6 82	7.50	7.97	8.64
Lost by radiation ..... Q'_e....	2.45	2.48	2.50	2.51
Second intermediate receiver—				
Supplied by jacket..... Q'_{jR} ..	4.20	4.04	4.27	4.22
Lost by radiation ..... Q'_{eR}...	1.20	1.22	1.23	1.24
Low-pressure cylinder—				
Brought in by steam... Q''....	132.14	130.50	138.61	147.33
During admission..... Q''_a...	5.85	3.05	5.57	5.29
During expansion..... Q''_b...	−9.51	−7.09	−8.65	−10.13
During exhaust..... Q''_c....	2.53	2.23	−1.44	−0.11
During compression.... Q''_d...	0.00	0.00	0.00	0.00
Supplied by jacket..... Q''_j....	7.08	6.20	7.41	7.14
Lost by radiation ..... Q''_e....	4.34	4.40	4.45	4.47
Total loss by radiation:				
By preliminary test.... ΣQ_e ..	10.07	10.20	10.31	10.35
By equation (49).....	11.68	10.19	8.75	8.07
Power and economy:				
Heat equivalents of works per stroke:				
High-pressure cylinder. $A\ W$..	8.44	8.34	9.17	9.52
Intermediate cylinder.. $A\ W'$.	7.12	6.95	7.77	8.42
Low-pressure cylinder.. $A\ W''$.	9.64	10.06	10.87	11.79
Totals.....	25.20	25.35	27.81	29.78
Total heat furnished by jackets .	27.58	27.02	27.71	28.45
Distribution of work:				
High-pressure cylinder	1.00	1.00	1.00	1.00
Intermediate cylinder.....	0.84	0.83	0.85	0.88
Low-pressure cylinder.....	1.14	1.21	1.19	1.24
Horse-power	104.9	104.2	113.1	120.3
Steam per horse-power per hour.	14.64	14.31	13.90	13.73
B. T. U. per horse-power per minute	247	241	236	232

DISCUSSION.

Mr. R. S. Hale.—There is one point which is hardly touched on in Professor Thurston's paper, and that is the question of omitting one or more jackets from a compound engine. It is probably standard practice to jacket the receivers of a compound or triple if the cylinders are jacketed. But Professor Peabody seems to have found otherwise in his trials at the Massachusetts Insti-

tute of Technology. On page 841 Professor Thurston states: "The best trial reported gave . . . no receiver jackets being in operation." Professor Wity, of Lille, reports in *La Revue générale des Sciences*, April 15, 1893, a similar result, as shown by the following table:

	Toutes les enveloppes chauffées.	Enveloppes I et II seules chauffées.	Enveloppes I et R seules chauffées.	Enveloppe R seule chauffée.	Aucune enveloppe chauffée.
Pression aux chaudières	6k.32	6k.27	6k.28	6k.20	6k.25
Température du condenseur	30°.9	30°.5	32°.0	32°.1	31°.5
Travail indiqué en chevaux	547,834	552.618	543,704	556,618	350,276
Purges des enveloppes I	1.75%	1.75%	1.66%	—	—
Purges des enveloppes II	2.12%	3.56%	—	—	—
Purges des enveloppes R	8,22%	—	8.34%	9.08%	—
Consommation brute par cheval-heure indiqué	6k.302	6k.181	6k.450	6k.614	6k.547
Consommation nette après déduction des calories des purges réintégrées aux chaudières	6k.107	6k.097	6k.286	6k.461	6k.547
Consommation sur piston	5k.540	5k.852	5k.805	6k.013	6k.547
Admission moyenne au petit cylindre	0.181	0.193	0.188	0.189	0.207
Condensation à l'admission	14%	14%	15%	18%	17%
Phénomène en détente	évap.=3.2%	évap.=3.5%	évap.=1.1%	cond.=1.6%	cond.=7.0%
Rapport du travail absolu ou travail disponible	0.688	0.633	0.580	0.626	0.584

NOTE.— En marche, le purgeur au receiver ne donne point d'eau.

In jacketing receivers, it seems to have been forgotten that the use of the jacket is chiefly to prevent initial condensation, which does not or should not occur in a receiver to any marked degree. The receiver jacket does give the next cylinder dryer steam, which is an advantage, but it could be just as easily obtained by a separating device in the receiver, while every thermal unit transferred from the jacket to the working steam is used at a lower temperature and hence at a lower efficiency than if used from boiler pressure and temperature.

Mr. Albert F. Hall.—It might be well to state that the Pawtucket pumping engine, with which Professor Denton made his experiments, cannot, strictly speaking, be run without steam jackets, since *the jackets of the heads of neither high nor low pressure cylinders can be thrown out of action.*

Mr. George I. Rockwood.—An interesting question is raised by Professor Thurston, in the statement on the first page of the introductory portion of his paper, as follows: "It was, however, found that there existed a substantial unanimity of opinion among those best qualified to express their views, whether from the standpoint of experience, as derived by long periods of occupation in designing and constructing by engineers whose reputation had been built up by years of success in construction, or from that of a man of science, and by the experimental investigation of the action of the steam-jacket under the various conditions of usual practice."

The question raised is: Who are the particular men, of all others best qualified to express their views on the action of steam in a modern engine? On the answer to this interesting question depends the truth or error of the general statement quoted above.

To begin with, we note a difference of opinion acknowledged in the foot-note of page 791 of the paper, where it is said Cotterill considers that to be the real use which the author considers only a secondary effect of the jacket. Is not Cotterill also competent to express his views? Take another instance of broad disagreement as to the effect of jacketing. It is that between those, on the one hand, who believe the principal use of the jacket that of prevention of original drops of moisture forming on the walls of the clearance spaces and cylinder just before admission of steam, on the theory that the incoming steam will condense in much larger quantities with moisture present than with it absent, and those, on the other hand, whose experiments with initially dry and initially wet surfaces do not reveal the occurrence of any more condensation due to the presence of the moisture than will occur without it. The late Peter Willans was a supporter of the first view, and Professor Denton is a supporter of the second.

Notwithstanding this, is not Willans or Denton competent to express a proper view of jacket action, whether from the standpoint of experience or of science? Again, Bryan Donkin found that jacketing a triple-expansion mill engine of English construction gave a large percentage of net gain, even not reckoning the jacket water as returnable to the boiler; while Denton found a gain of almost insignificant value in the use of jackets on a triple-expansion pumping engine built in this country. Perhaps it cannot be denied that all these experimenters are competent to have views on this subject worthy of our attention. Indeed, on second thought, to whom, besides Professor Thurston, can we point in the

whole civilized world as having views entitled to greater confidence than the transatlantic opinions of Donkin, or the cisatlantic opinions of Denton? Yet are they non-concordant. What are we dimmer flames to do, then? Are we to accept the dogmatic assertions of the author, to the effect that all proper authorities support substantially the same views, namely, those of James Watt? If so, then why the necessity, at the close of this century of familiarity with the steam-jacket, of the lengthy arguments given by the author in the paper?

I ask forgiveness for devoting this time to this opening remark of the author, on the ground that, so far as I can judge, it is the only portion which heretofore has not been passed and repassed through the mangle of public scrutiny, and because the paper which follows the remark is not especially corroborative of it. The large array of collated data may be made to support the conclusions he quotes from the pocket-book of Messrs. Seaton and Roundthwaite, or to thoroughly undermine them—according to the stress laid upon the value of different portions of the data.*

We must face the fact that the instruments, methods, and ideas of the present date are so far advanced and refined, as contrasted with those of even ten years ago, that vast aggregations of data collected previous to that time are not convincing when taken as the basis of a present dogmatic theory. Before giving one's self to the luxury of formulating one—not to speak of a dozen, to suit a dozen different hypothetical cases—it is best to wait, to go on testing, to accumulate data in regard to the accuracy of which there is no ground for disbelief, data which everybody has confidence in. In these days fulminations of "authorities" will not do in addressing "authorities"; correct theory, based on indisputable facts alone, is of value.

In the meantime we can have our private views, but can we put them forth as bearing the stamp of authority, as views which all must accept or be *anathema maranatha?*

Prof. R. H. Thurston.—Mr. Hale brings up an important and interesting matter, which, however, I think we are hardly yet

* Observe (the first sentence of the first paragraph on page 843) the easy conviction expressed, that since we have this enormously irregular and unaccountable variation of results, therefore the only conclusion derivable is that the steam-jacket may be made efficient under ALL conditions of operation of the steam-engine. Ought, or ought not, this to be characterized as a case of *non sequitur?*

prepared to settle, not having as yet sufficient data. There is no question as to the nature or effect of the action of the receiver-jacket; but there remains to be determined by investigation just how its immediate action affects the economic performance of the machine as a whole. General experience seems to have led builders to the use of this jacket, and the presumption would ordinarily be accepted that they must have thus secured improved efficiencies. The receiver-jacket is a jacket on a separator; the purpose of which separator is simply to remove from the passing current of steam the water of condensation due to external radiation and thermodynamic conversion of heat into work. If it is efficient as a separator, the succeeding cylinder will receive dry steam. In so far as it is inefficient in this separation, the jacket has a work to perform in the further drying, possibly even the superheating, of the wet steam. This action is well shown in the diagrams from the Sibley College experimental engine, designed under the direction of the writer, and with jackets on all cylinders and receivers, as here operated. (Fig. 249.)

In this case, the quality of the steam varies from an average of about 0.75 in the first, to over 0.90 in the second, and to 1.10 in the third cylinder; becoming steadily dryer and finally superheated by the action, mainly, of the receiver-jackets; which latter fact is shown by the progressive falling off of quality in each cylinder, after entrance, from 0.77 to 0.72 in the first, from 0.92 to 0.88 in the second, and from 1.20 to unity in the third cylinder. In this particular instance the result was economically exceptional; the consumption of steam being the lowest recorded, so far as known, from an engine of this type, and at such low pressure.* What would be the result of similar jacketing on other engines cannot be predicted; but the interesting and instructive fact is here well brought out, that a receiver-jacket may produce superheating, and that this superheating may make the jacketing of the succeeding cylinder superfluous. The same fact, already fairly well brought out, both in the present and in the preceding introductory paper, is also illustrated by the behavior of the engines of the steamships *Hudson* and *Knickerbocker* of the line between New York and New Orleans, in which the jackets were found to

* These facts are extracted from a paper prepared by the writer for this meeting of the Society, but withheld for the time, awaiting additional matter kindly offered by Professor Carpenter relating to its subject: "Method of Initial Condensation and Heat-Wastes in Steam-Engine Cylinders."

have no effect when the superheating amounted to 65° Fahr.* This supersedure of jacketing by superheating is also again illustrated in the paper presented at this meeting on "The Compound Harris-Corliss Engine," in which the use of steam superheated only 15° Fahr. sufficed to reduce cylinder condensation to 12.5 per cent., and steam consumption to 15.5 pounds per horse-power per hour. †

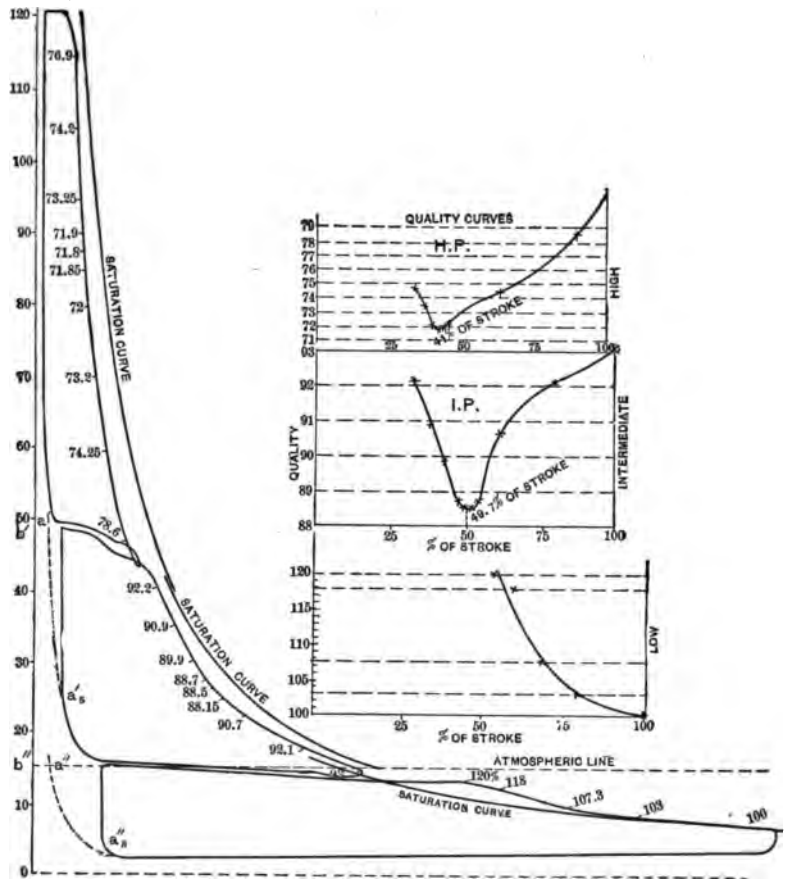

FIG. 249.—Steam Jacketing Complete.

The following are the particulars of the performance of the Sibley College engine, so far as here of importance, as obtained by the trial giving the mean diagram, which has been discussed above:

* "Authorities on the Steam-Jacket:" *Trans. A. S. M. E.*, 1890, Vol. XII., p. 507.
† No. DC.; *Trans.* 1894, Vol. XV.

Cylinders, 9 + 16 + 24 × 36 inches.	Clearances, 7.6, 8.93, and 9.35%.
Piston rods, 2.31 inches diameter.	Boiler pressure, 125 (abs.); 110 by gauge.
Vacuum, 10.8 lbs., 22 inches.	Barometer, 29.4, 14.42 inches.
Jacket water, 13.72%.	Condensing water, 19 lbs. per lb. steam.
Total I.H.P., 140.2.	D.H.P., 123.4.
Mechanical efficiency, 0.88.	Steam, 13.3 lbs. per I.H.P. per hour.
B.T.U., per I.H.P. per hour, 14,160.	Steam, 15.1 lbs. per D.H.P. per hour.
B.T.U., per I.H.P. per min., 236.	Total ratio of expansion, 13.83.

Pressures (abs.), at cut-off, 131, 43, 13.5; at release, 43, 14.5, 2.5.

Jacket water, per cent., 26.4, 7.05, 28.1 in cylinders, and 9.85, 34.6, in receivers.

Work per cent., 1, 1.33, and 1.675 in cylinders 1, 2, and 3.

Thermodynamic efficiency of Carnot cycle, 24.7%; actual, 18%; ratio of actual to Carnot, 0.73.

Water rate of ideal Rankine cycle, 9.6 lbs.; ratio to actual, 0.72.

This fact, as indicated in the body of this paper and also in the first paper on the same subject, that every condition reducing the internal wastes correspondingly reduces the necessity of jacketing and annuls its action, is exhibited in the tables on pages 814 and 819 of the paper on the multiple-cylinder engine,* in which the reduction of the initial condensation from twenty to thirty per cent. with saturated or wet steam, to from eighteen to twenty-five by a few degrees of superheat, to from thirteen to eighteen by twenty-four to thirty degrees, and to five and ten by sixty to eighty degrees, is shown. Thus, even a very moderate degree of superheating accomplishes as much, usually, as a jacket, under similar conditions in other respects, and to that extent supersedes its use. Thus the receiver jacket may prove of real advantage, and may, to this extent, make the engine jacket—at least on the low-pressure cylinder—unnecessary; as may any device which checks those wastes which it is intended to reduce. Very moderate superheating may thus make the steam-jacket superfluous or comparatively inactive.

Mr. Hall's remark may throw some light upon the apparent singular inefficiency of the jackets of the Pawtucket pumping engine, which seems to stand almost alone in that respect among engines of that class. The work of the jacket on the cylinder body is performed by other processes of reduction of wastes or by the jacketed heads, leaving little for it to do. Jackets on the heads are now well known to be vastly more useful and efficient than on the sides; where, in fact, they are comparatively at a great

* No. CCCLXII., *Transactions of American Society of Mechanical Engineers*, Vol. XI.

disadvantage in combating a waste which occurs almost entirely on cylinder heads and pistons.

It will be noticed that, in the case of the Sibley College engine above referred to, the steam is precisely dry at exhaust when the engine is doing its best recorded duty—to date. This corresponds with the maxim of Ledieu, and of Professor Dwelshauvers-Dery, who says: "Maximum efficiency is obtained when the steam is dry at the beginning of exhaust." * The same investigator corroborates, by his latest work, published since this paper was submitted, my deduction of a jacket-action of maximum efficiency. He finds his own experimental engine to justify these conclusions: †

"Steam-jacketing in this engine gives greater economy at $\frac{1}{10}$ cut-off than at $\frac{3}{10}$. Short cut-offs are not economical without the use of the jacket. Thus, from $\frac{1}{10}$ to $\frac{3}{10}$, there is a gain of 4.84 %, while *with* jackets this result is reversed, and there is a gain of 3.7% in diminishing the cut-off from $\frac{3}{10}$ to $\frac{1}{10}$."

The exact point of maximum effect is, of course, variable with every variation of conditions affecting final economy.

Since this paper was presented I have also received the data and the results of trial of the jacketed Louisville compound pumping engine designed by Mr. E. D. Leavitt, which are the following:

The steam cylinders are 27 and 54 inches in diameter, 10 feet stroke of piston; working two differential plunger pumps of diameters $24\frac{1}{16}$ and 34 inches and of 7 feet stroke. The trial was of 144 hours length, "as the chief engineer and superintendent do not believe in '*truncated*' *tests*." The coal was Pittsburg for three days, Pocahontas the second three days; neither being rated as of best quality. The dynamic head was 183.6 feet, and a duty was obtained of 129,828,151 ft.-lbs. per 100 lbs. of fuel, as figured by the reporters. The mechanical efficiency was 93.4%; friction being but 6.6%. The consumption of dry steam amounted to 12.7 lbs. per horse-power per hour. The steam condensed in jackets and reheaters amounted to 16.7% of the total; that in the jackets alone was not determinable. The working pressure was 140 lbs. Reducing this duty to the new unit, 1,000 lbs. dry steam, it becomes, in round numbers, 152,000,000. For a compound

* *Engine Trials at the University of Liège; Étude Expérimentale Calorimétrique de la Machine à Vapeur*, p. 75. *Manual of the Steam Engine:* R. H. T., Vol. I., pp. 648, *et seq.*

† *Revue Universelle des Mines*, T. XXV., p. 324, 1894. *Sibley Journal of Engineering*, June, 1894.

engine this is a very remarkable figure, and it may well be imagined that the use of the jacket was in some degree contributive to its success.

In closing this discussion, I should feel that I ought to apologize for the space taken up in the treatment of the subject of steam-jackets in these two papers on "Current Opinion" and on "Current Practice," but that the importance of the subject and of the question involved is fully commensurate with their extent. It will be noted that the importance of the steam-jacket may be rated, for some classes of engines, at least, as from ten to twenty per cent. of that of the engine itself, efficiency being the gauge; and unnumbered volumes already written are insufficient to cover the latter, while this long-neglected but important detail has received comparatively little attention in the literature of the steam engine in either America or Great Britain, although it finds its proper place and deserved attention in works of Continental writers. I have endeavored to collate the views of the acknowledged authorities among designers, builders, users, and investigators throughout the world, and think that this compilation will be found by those interested fairly full and yet free from unimportant or irrelevant matter in any large amount. In some cases, to make each paper complete and symmetrical from its special point of view, some repetition has been permitted. This may also be justified, I doubt not, by the Scripture admonition: "Line upon line and precept upon precept"; and it may still be possible that there may be found those who believe that under no circumstances can the jacket be profitably employed at speeds of revolution exceeding fifty; that it would be reasonable to expect that jacketing should be as effective at high as at low speed, with superheated as with wet steam; or that a saving of three or five per cent. will never justify the use of the jacket. But I am sure that no one will claim that the discovery of an inefficient jacket disproves the value of the jacket, any more than the discovery of a leaky engine disproves the desirability of making pistons tight. The two papers, however, will be found, I think, to furnish substantially all available evidence to date, and every member of the profession will make his own deductions as to the method of action of the jacket, the causes and limitations of its beneficial influence, and the advisability, from a financial point of view, of adopting it in any specified case or for any given class of engine.

DXCI.*

RESULTS OF MEASUREMENTS OF THE WATER CONSUMPTION OF AN UNJACKETED 1600 HORSE POWER COMPOUND HARRIS-CORLISS ENGINE.

BY J. E. DENTON, D. S. JACOBUS AND R. H. RICE.

(Members of the Society).

THE engine under notice furnishes the motive power for the Bristol Manufacturing Company's fifty thousand Spindle Cotton Mill at New Bedford, Mass. It is of the cross-compound type, and of the following dimensions:—

Diameter high-pressure cylinder	30.025 inches.
Diameter low-pressure cylinder	55.915 inches.
Stroke of both cylinders	72.00 inches.
Average clearance each end high-pressure cylinder, per cent. piston displacement	2.6 per cent.
Average clearance each end low-pressure cylinder, per cent. piston displacement	3.6 per cent.

Each cylinder has separate eccentrics for the admission and exhaust valves, respectively. The cylinders are unjacketed on both the head and barrels.

The receiver between the cylinders is a cylinder 29 inches diameter and 13 feet long, having an annular space about its barrel which was intended to be used as a live-steam jacket. This jacket is not generally used. In the experiments here described no live steam was admitted to it, and it was connected with the interior of the receiver; all drip pipes for draining the jacket, and the interior of the receiver, being tightly closed.

The feed water was measured by a carefully calibrated meter placed between the feed pumps and the boilers.

All of the steam generated by the boilers passed through the engine, except a small portion amounting to 165 lbs. per hour, which represented the combined leakage from the blow-off cocks

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Vol. XV. of the *Transactions*.

and water-glass drips; 93 lbs. per hour, which was the average amount used by a Barrus calorimeter; and a portion amounting to 230 lbs. per hour, which was drawn from a dead end in the live-steam pipe leading to the low-pressure cylinder.

The first quantity was determined by collecting and weighing the leakage during a certain time. The second quantity was determined by calibrating the orifice of the calorimeter. The latter quantity was determined by the calculation given on page 886.

The detailed data of the test are shown in Tables I. to XIII. The results show a water consumption of 13.50 lbs. of steam per hour per indicated horse power with 125 lbs. boiler pressure, 0.30 cut-off in high cylinder, steam 14.6 degrees of superheated at the throttle-valve, and 65.2 revolutions of the engine per minute.

The feed water for the engine was drawn from the city supply at 40° Fahr., and passed through a nest of tubes, aggregating 400 square feet of surface, lodged in a cylinder through which the exhaust steam was discharged into the condenser. By means of this exhaust heater the temperature of the feed water was raised to 131.3°. The feed water then passed through a drip tank, in which it was mixed with the steam from the dead end of the steam pipe, whereby its temperature was raised to 142.7°.

In the ordinary operation of the mill the condensed steam from the slashers, etc., also mixes with the feed water in the drip tank, whereby a temperature upwards of 160° Fahr. is given to the feed water. At this temperature of feed water the boilers, which were of the Bigelow-Manning type, evaporate $10\frac{1}{3}$ lbs. of water per pound of Pocahontas bituminous coal.

The plant as a whole, therefore, affords the remarkable economy of $1\frac{1}{3}$ lbs. of coal per indicated horse power per hour, with an unjacketed compound engine expanding steam about thirteen times.

The writers are indebted to the courtesy of Mr. Chas. C. Diman, superintendent of the mill, for the opportunity to test the engine, and for his cordial co-operation in dispensing with the use of live steam in the mill, during the period of the measurements, for all purposes except the operation of the engine.

CALCULATION OF STEAM CONSUMED PER HOUR, AND PER HOUR PER HORSE POWER.

The total weight of water registered by the meter in four and one-half hours was 1529.1 × 64.70 = 98933 lbs., or 21985 lbs. per hour. From this there is to be deducted the leakage of the blow-off valves of the boilers, which amounted to 165 lbs. per hour, the steam which passed through the Barrus calorimeter, and the steam which flowed from a dead end in one of the steam pipes in order to prevent water collecting in the same. The amount of steam returned from the dead end is calculated from the rise in temperature of the feed water, assuming that the steam is dry. This will give too low a correction on account of the amount of moisture that may be present in the steam, but as the whole correction is a small one the error involved is not of importance.

The error involved by neglecting any moisture contained in the steam thus returned to the feed water will be against the economy of the engine.

The increase of temperature of the feed water due to the returned steam was 142.7 − 131.3 = 11.4° Fahr. The heat units imparted to the feed water per hour will be 11.4 (21985 − x) in which x is the amount of steam condensed. Each pound of steam parts with 1221 − 142.7 = 1078.3 B. T. U., so that we have 1078.3 x = 11.4 (21985 − x), from which obtain x = 230 lbs. per hour.

The amount of steam passing through the Barrus calorimeter was 102 lbs. per hour. The calorimeter was in operation four hours and five minutes, so that the total steam passing through it was 416.5 lbs., or the average rate was 416.5 ÷ $4\frac{1}{2}$ = 93 lbs. per hour.

The total amount to be deducted from the water passing through the meter is, therefore, 165 + 230 + 93 = 488 lbs., and the net steam consumed by the engine per hour is 21985 − 488 = 21497 lbs.

The steam per hour per horse power is 21497 ÷ 1592.2 = 13.50 lbs.

CALIBRATION OF INDICATOR SPRINGS.

The standard of pressure for the springs used on the high pressure cylinder was the Utica Steam Gauge Company's

weight device, which, as was stated in the paper on the Comparison of Indicators presented at the last meeting of this Society, was found to agree with the mercury column at the Brooklyn Navy Yard.

The standard of pressure used for the low pressure indicators was a mercury column, the readings of which were verified by a distilled-water column in order to make sure that the density of the mercury, which was the ordinary mercury of commerce, did not vary from the standard figure.

The readings of the mercury column agreed precisely with the readings of the distilled-water column, so that the density of the commercial mercury was the same as for chemically-pure mercury.

The same mercury was employed in the column used for measuring the vacuum in the engine test as was employed in the tests for standardizing the springs.

The general method of standardizing the indicators, and calculation of the equivalent scales of the springs, so as to allow for all variations in the scales at different heights on the diagrams, is the same as was given in detail in the paper on the Comparison of Indicators already mentioned.

The results of tests of the springs are given in detail in Tables VII. and VIII., and the calculation of the equivalent scales in Tables IX. to XIII.

TABLE I.

FINAL RESULTS OF TEST.

Horse power			1592.2
Steam per hour per horse power, lbs			13.50
Average pressure at engine throttle, lbs. per square inch above atmosphere			128.0
Average superheating at engine throttle, degrs. Fahr			14.6
Average vacuum, inches of mercury			25.6
Lbs. of steam per hour per horse power calculated from indicator cards.	High pressure cylinder.	Near cut-off	11.81
		Near release	12.02
	Low pressure cylinder.	Near cut-off	10.77
		Near release	11.41
Steam not accounted for at cut-off of high pressure cylinder, per cent			12.5
Revolutions per minute			65.21
Piston speed in feet per minute			783
Ratio of expansion			13.4
Ratio of actual mean effective pressure to mean effective pressure for Marriotte curve and two pounds back pressure			0.79

AVERAGE INDICATOR DIAGRAMS.

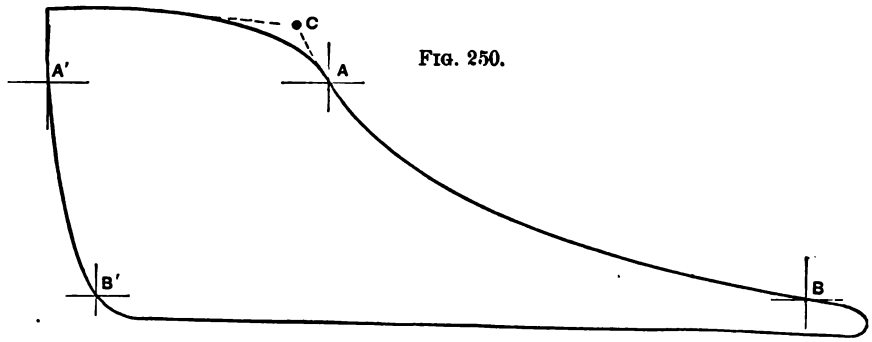

FIG. 250.

FIG. 250.—Head end of high-pressure cylinder. The steam accounted for by indicator is calculated for the net volumes AA' and BB'.

Spring D.—Equivalent scale, 60.29. M.E.P., 48.55. Horse-power, 407.6.

FIG. 251.

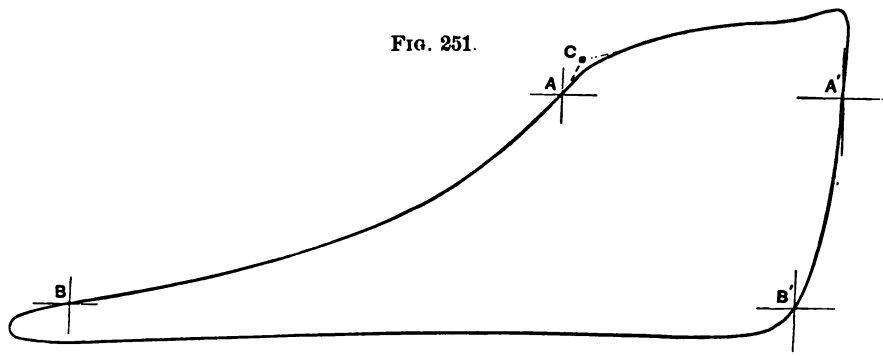

FIG. 251.—Crank end of high-pressure cylinder. The steam accounted for by indicator is calculated for the net volumes AA' and BB'.

Spring C.—Equivalent scale, 60.13. M.E.P., 48.64. Horse-power, 397.6.

FIG. 252.

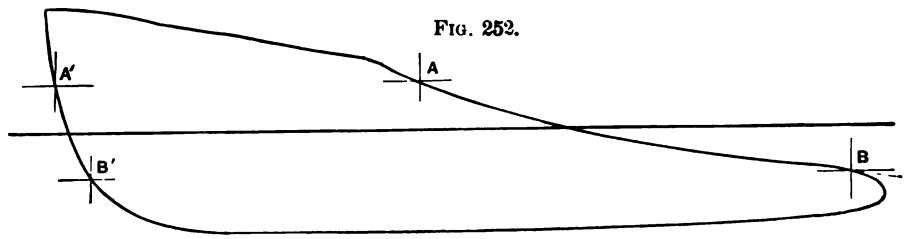

FIG. 252.—Head end of low-pressure cylinder. The steam accounted for by indicator is calculated for the net volumes AA' and BB'.

Spring A.—Equivalent scale, 20.61. M.E.P., 13.65. Horse-power, 397.4.

FIG. 253.

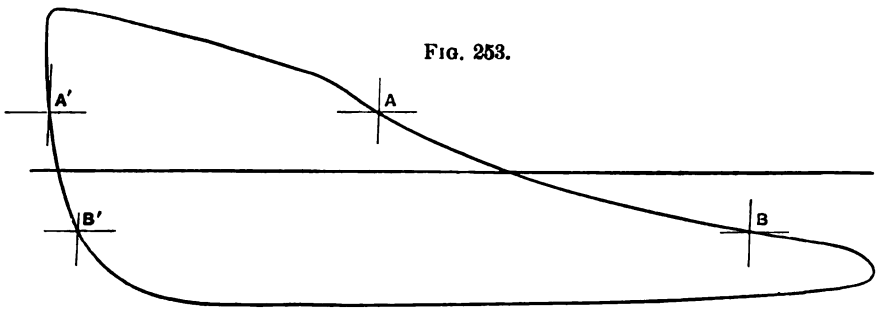

FIG. 253.—Crank end of low-pressure cylinder. The steam accounted for by indicator is calculated for the net volumes AA' and BB'.

Spring B.—Equivalent scale, 16.74. M.E.P., 13.50. Horse-power, 389.6.

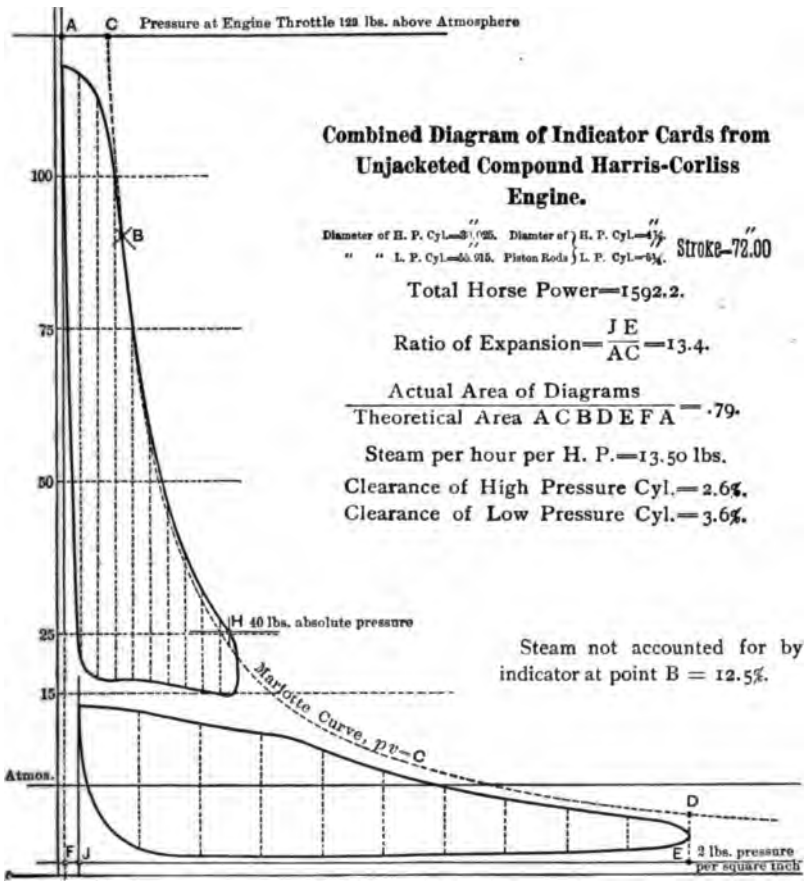

FIG. 254.

TABLE II.

NET AREAS OF CYLINDERS AND HORSE-POWERS.

	Net area in square inches.	Stroke × area × revolutions ÷ 33,000.	Horse-power.
High-pressure cylinder, head end	708.04	8.3948	407.6
High-pressure cylinder, crank end	689.37	8.1735	397.6
Average and total	698.70	.....	805.2
Low-pressure cylinder, head end	2,455.54	29.1188	327.4
Low-pressure cylinder, crank end	2,433.89	28.8571	389.6
Average and total	2,444.71	.....	787.0

Ratio of volume of low-pressure cylinder to volume of high-pressure cylinder = 3.50.
Total horse-power = 805.2 + 787.0 = 1,592.2.

TABLE III.

DATA TAKEN AT ENGINE.

Time. P. M.	Revolution counter.	Steam Pressure in Lbs. per Square Inch above Atmosphere.		Vacuum in inches of mercury by mercury column.	Temperature of Steam 3 Ft. from Engine, in Deg. Fahr.		Superheating in degrees Fahr.
		In main steam pipe 3 ft. from engine.	Receiver.		Actual reading of thermometer.	Reading of thermometer for satur. steam	
1.00	936,536	123.5	14.0	25.4	363	348	15
1.30	938,492	123	13.5	25.7	362	348	14
2.00	940,450	123	13.3	25.7	363	348	15
2.30	942,406	122	13.5	25.7	363	347	16
3.00	944,362	123	13.3	25.7	365	348	17
3.30	946,319	123	13.4	25.7	363	348	15
4.00	948,277	123	13.5	25.7	361	348	13
4.30	950,233	123.5	13.6	25.4	362	348	14
5.00	952,188	123	13.7	25.4	361	348	13
5.30	954,143	123.5	13.4	25.3	362	348	14
Per minute and average.	65.21	123.0	13.5	25.6	362.5	347.9	14.6

TABLE IV.

DATA TAKEN AT BOILERS.

Time, P.M.	Meter readings.	Temperature at meter.	Temperature at drip-tank.	Gauge on boiler No. 1.	Water Levels, in Inches.									Corrected meter readings.*	Half-hourly rates by meter.
					No. 8.	No. 7.	No. 6.	No. 5.	No. 4.	No. 3.	No. 2.	No. 1.	Sum.		
1.00	31,905	147	131	125	16.0	10.0	11.8	12.5	7.5	9.5	11.5	8.3	87.1	31,905	.....
1.30	32,084	142	129	124	15.0	12.5	12.5	14.5	8.5	10.5	11.5	7.8	92.8	32,078.5	173.5
2.00	32,243	141	129	125	10.0	11.3	12.5	11.0	8.0	10.5	11.0	8.0	82.3	32,247.6	169.1
2.30	32,425	141.5	131	124	12.0	11.0	12.0	12.5	12.0	11.5	13.8	9.5	94.3	32,418.1	170.5
3.00	32,576	143.5	131.5	125	12.0	7.0	11.5	9.5	11.5	10.8	8.0	7.2	77.5	32,585.2	167.1
3.30	32,751	139.5	129.5	126	12.5	10.0	9.8	9.0	10.0	13.0	11.5	9.0	84.8	32,753.2	168.0
4.00	32,926	141.5	131.5	126	11.3	12.0	12.0	12.0	13.2	10.2	11.0	9.8	91.5	32,921.8	168.6
4.30	33,094	143	133	126	10.0	11.5	11.2	12.0	12.2	12.0	11.5	9.5	89.9	33,091.3	169.5
5.00	33,262	143	133	126	10.5	10.5	15.0	9.0	12.0	9.0	10.5	8.5	85.0	33,264.0	172.7
5.30	33,429	145	134.5	126	11.2	8.4	13.2	8.0	11.2	11.0	11.0	7.8	81.8	33,434.1	170.1
	Average	142.7	131.3	125.3									86.7	Total	1,529.1

* One inch height in boiler = 0.96 cubic foot registered by meter.

TABLE V.

MEAN EFFECTIVE PRESSURES AND HORSE-POWERS.

Time. P.M.	High-Pressure Cylinder—Head.					High-Pressure Cylinder—Crank.					Low-Pressure Cylinder—Head.					Low-Pressure Cylinder—Crank.				
	Mean Height of Diagram.			Scale of spring.	Mean effective pressure.	Mean Height of Diagram.			Scale of spring.	Mean effective pressure.	Mean Height of Diagram.			Scale of spring.	Mean effective pressure.	Mean Height of Diagram.			Scale of spring.	Mean effective pressure.
	Computer No. 1.	Computer No. 2.	Average.			Computer No. 1.	Computer No. 2.	Average.			Computer No. 1.	Computer No. 2.	Average.			Computer No. 1.	Computer No. 2.	Average.		
1:05	0.8333	0.8321	0.8327	60.29	50.20	0.8218	0.8225	0.8222	60.13	49.44	0.8404	0.8380	0.8392	16.73	14.04	0.6609	0.6621	0.6615	20.60	13.63
1:30	0.7724	0.7697	0.7710	60.29	46.48	0.8034	0.8019	0.8026	60.13	48.26	0.8365	0.8341	0.8353	16.73	13.97	0.6578	0.6582	0.6580	20.60	13.55
2:00	0.8089	0.8089	0.8089	60.29	48.77	0.7962	0.7947	0.7954	60.13	47.83	0.8166	0.8170	0.8168	16.73	13.67	0.6540	0.6544	0.6542	20.60	13.48
2:30	0.8096	0.8067	0.8082	60.29	48.73	0.8056	0.8063	0.8060	60.13	48.46	0.8201	0.8209	0.8205	16.73	13.73	0.6545	0.6568	0.6557	20.60	13.51
3:00	0.8364	0.8341	0.8352	58.89	49.18	0.8088	0.8069	0.8079	60.13	48.58	0.8221	0.8189	0.8205	16.73	13.73	0.6551	0.6522	0.6537	20.60	13.47
3:30	0.8400	0.8356	0.8378	58.89	49.34	0.8142	0.8111	0.8127	60.13	48.87	0.6419	0.6430	0.6425	20.61	13.24	0.8002	0.8006	0.8004	16.74	13.40
4:00	0.8119	0.8100	0.8110	58.89	47.76	0.8131	0.8108	0.8119	60.13	48.82	0.6560	0.6545	0.6552	20.61	13.50	0.8084	0.8069	0.8076	16.74	13.52
4:30	0.8123	0.8107	0.8115	58.89	47.79	0.8119	0.8096	0.8108	60.13	48.75	0.6552	0.6557	0.6560	20.61	13.52	0.8087	0.8082	0.8084	16.74	13.45
5:00	0.7939	0.7951	0.7945	58.89	46.79	0.7900	0.7877	0.7888	60.13	47.43	0.6644	0.6613	0.6628	20.61	13.66	0.8060	0.8037	0.8049	16.74	13.47
5:30	0.8586	0.8551	0.8569	58.89	50.46	0.8321	0.8299	0.8310	60.13	49.97	0.6520	0.6534	0.6527	20.61	13.45	0.8057	0.8063	0.8060	16.74	13.49
Average					48.55	Average.....				48.64	Average.....				13.65	Average.....				13.50
Horse power.....					407.6	Horse power				397.6	Horse power				397.4	Horse power.....				389.6

TABLE VI.

CALCULATION OF STEAM ACCOUNTED FOR BY INDICATOR CARDS.

Time.	Cut-off in High-Pressure Cylinder in per cent. of Stroke.*		Ratio of net volumes of cylinders filled with steam at the pressures given below.							
			High-pressure Cylinder.				Low-pressure Cylinder.			
			Near Point of Cut-off at 105 lbs. Absolute Pressure.		Near Point of Release at 40 lbs. Absolute Pressure.		Near Point of Cut-off at 20 lbs. Absolute Pressure.		Near Point of Release at 10 lbs. Absolute Pressure.	
P. M.	Head End.	Crank End.	Head End.	Crank End.	Head End.	Crank End.	Head End.	Crank End.	Head End.	Crank End.
1:05	0.307	0.306	0.355	0.356	0.879	0.907	0.451	0.425	0.899	0.908
1:30	0.283	0.285	0.321	0.325	0.809	0.854	0.437	0.412	0.890	0.832
2:00	9.296	0.290	0.350	0.328	0.882	0.857	0.424	0.418	0.883	0.846
2:30	0.297	0.301	0.345	0.340	0.871	0.884	0.428	0.410	0.879	0.822
3:00	0 304	0.298	0.358	0.340	0.900	0.866	0.420	0.423	0.872	0.892
3:30	0.301	0.289	0.349	0.337	0.862	0.890	0.435	0.402	0.915	0.788
4:00	0.299	0.294	0.346	0.339	0.860	0.853	0.432	0.400	0.892	0.805
4:30	0.308	0.307	0.341	0.338	0.850	0.846	0.448	0.402	0.910	0.788
5:00	0.302	0.283	0.334	0.325	0.834	0.821	0.445	0.402	0.911	0.801
5:30	0.322	0.312	0.361	0.349	0.921	0.860	0.448	0.411	0.915	0.827
Average	0.302	0.296	0.346	0.328	0.867	0.858	0.437	0.410	0.897	0.829
	0.299		0.342		0.862		0.424		0.863	

Ratio of Expansion
- From point marked C in Figs. 1 and 2 ..... 11.7
- From the initial pressure point C, Fig. 5 ..... 13.4

High-Pressure Cylinder.
- Average mean effective pressure in lbs. per square inch 48.59
- Equivalent M. E. P. of low-pressure cylinder, 13.57 × 3.50 ..... 47.49
- Total M. E. P. reduced to high-pressure cylinder 96.08
- Steam per hour per horse-power = Ratio × density × 13.750 ÷ total M. E. P.
  - Near cut-off $= \frac{0.342 \times 0.2414 \times 13750}{96.08} =$ 11.81
  - Near release $= \frac{0.862 \times 0.0974 \times 13750}{96.08} =$ 12.02
- Steam not accounted for by indicator
  - Near cut-off 12.5%
  - Near release 11.0%

Low-Pressure Cylinder.
- Average mean effective pressure in lbs. per square inch 13.57
- Equivalent M. E. P. of high-pressure cylinder, 48.59 × 3.50 ..... 13.88
- Total M. E. P. reduced to low-pressure cylinder 27.45
- Steam per hour per horse power = Ratio × density × 13.750 ÷ total M. E. P.
  - Near cut-off $= \frac{0.424 \times 0.0507 \times 13750}{27.45} =$ 10.77
  - Near release $= \frac{0.863 \times 0.0264 \times 13750}{27.45} =$ 11.41
- Steam not accounted for by indicator
  - Near cut-off 20.2%
  - Near release 15.5%

* Apparent cut-off measured at points marked C, Figs. 1 and 2.

TABLE VII.

TESTS OF SPRINGS IN INDICATORS USED ON LOW-PRESSURE CYLINDER.

Spring.	Number of Card.	Heights measured on Diagrams, in Inches, from Atmospheric Line to Pressures given below.					
		Above Atmosphere.			Below Atmosphere.		
		5	10	15	4	8	12
A	1	0.245	0.500	0.750	0.190	0.385	0.570
A	2	0.245	0.505	0.750	0.190	0.380	0.575
A	3	0.250	0.495	0.745	0.195	0.380	0.575
A	4	0.250	0.500	0.750	0.190	0.385	0.575
A	5	0.250	0.500	0.745	0.195	0.385	0.575
Average		0.248	0.500	0.748	0.192	0.383	0.574
Scale		20.16	20.00	20.05	20.83	20.89	20.91
B	1	0.290	0.615	0.935	0.235	0.475	0.705
B	2	0.310	0.620	0.935	0.240	0.470	0.705
B	3	0.295	0.625	0.930	0.235	0.475	0.705
B	4	0.310	0.625	0.945	0.240	0.475	0.710
B	5	0.310	0.615	0.935	0.240	0.470	0.700
Average.....		0.303	0.620	0.936	0.238	0.473	0.705
Scale.....		16.50	16.13	16.03	16.81	16.91	17.02

TABLE VIII.

TESTS OF SPRINGS IN INDICATORS USED ON HIGH-PRESSURE CYLINDER.

Spring.	Number of Card.	Heights in inches measured on Diagrams starting from Lines 15 lbs. per square inch above the Atmosphere to Lines for Pressures given below.					Distance in inches from atmospheric line to lines for 15 lbs. gauge pressure.
		25	50	75	100	120	
C.....	1	0.165	0.585	1.000	1.405	.....	.....
C.....	2	0.165	0.585	1.005	1.415	.....	0.265
C.....	3	0.165	0.585	0.995	1.410	1.745	0.260
C.....	4	0.170	0.585	1.000	1.405	1.745	0.260
C.....	5	0.170	0.590	0.995	1.410	1.745	0.255
Average.....		0.167	0.586	0.999	1.409	1.745	0.260
Scale.....		59.88	59.73	60.06	60.33	60.17	.....
D.....	1	0.160	0.575	0.990	1.420	1.770	0.230
D.....	2	0.155	0.575	1.000	1.425	1.775	0.230
D.....	3	0.155	0.575	1.000	1.405	1.765	0.225
D.....	4	0.155	0.565	0.995	1.400	1.760	0.235
D.....	5	0.155	0.580	0.990	1.425	1.775	0.230
Average.....		0.156	0.574	0.995	1.415	1.769	0.226
Scale.....		64.10	60.98	60.30	60.07	59.36	.....
E.....	1	0.155	0.575	1.005	1.455	1.810	0.230
E.....	2	0.160	0.575	1.005	1.460	1.825	0.225
E.....	3	0.155	0.575	1.000	1.455	1.815	0.235
E.....	4	0.160	0.585	1.015	1.470	1.820	0.225
E.....	5	0.155	0.575	1.010	1.475	1.810	0.230
Average.....		0.157	0.577	1.007	1.463	1.816	0.229
Scale.....		63.69	60.66	59.58	58.10	57.82	.....

METHOD OF DETERMINING THE HEIGHT THAT THE POINTS AA′ AND BB′ ARE ABOVE OR BELOW THE ATMOSPHERIC LINE.

To find height for 105 lbs. absolute pressure. Spring D.

Difference of pressure between 105 lbs. absolute and 15 lbs. gauge pressure = 105 − 14.7 − 15 = 75.3 lbs.

Scale at 105 lbs. absolute pressure = 60.1 lbs. per inch.

Height that 105 lb. line is above the 15 lb. line = 75.3 ÷ 60.1 = 1.253″.

Height that 15 lb. line is above atm. line from Table VIII. = .226″.

Height that 105 lb. line is above atm. line. 1.48″.

TABLE IX.

Spring A.—Calculations of Equivalent Scale Corrected for all Variations in the Hot Scale of the Spring.

AVERAGE CARD, HEAD END OF LOW-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	+0.275	+0.629	20.03	+5.508	B1	−0.083	−0.190	20.83	−1.729
A2	+0.233	+0.533	20.01	+4.662	B2	−0.217	−0.497	20.90	−4.535
A3	+0.198	+0.453	20.04	+3.968	B3	−0.232	−0.531	20.91	−4.851
A4	+0.155	+0.355	20.09	+3.114	B4	−0.234	−0.536	20.91	−4.893
A5	+0.105	+0.240	20.16	+2.117	B5	−0.236	−0.540	20.91	−4.935
A6	+0.047	+0.108	20.16	+0.948	B6	−0.236	−0.549	20.91	−4.935
A7	−0.013	−0.030	20.83	−0.271	B7	−0.234	−0.536	20.91	−4.893
A8	−0.047	−0.108	20.83	−0.979	B8	−0.230	−0.526	20.90	−4.807
A9	−0.085	−0.195	20.83	−1.771	B9	−0.225	−0.515	20.90	−4.703
A10	−0.110	−0.252	20.85	−2.294	B10	−0.203	−0.465	20.90	−4.243
Totals...	+0.758	.....	.....	+15.002	Totals...	−2.130	.....	.....	−44.524

Equivalent scale = (15.002 + 44.524) ÷ (0.758 + 2.130) = 20.61.

AVERAGE CARD, CRANK END OF LOW-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	+0.303	+0.693	20.04	+6.072	B1	−0.119	−0.272	20.86	−2.482
A2	+0.255	+0.584	20.02	+5.105	B2	−0.229	−0.524	20.90	−4.786
A3	+0.223	+0.510	20.00	+4.460	B3	−0.233	−0.533	20.91	−4.872
A4	+0.163	+0.373	20.08	+3.273	B4	−0.233	−0.533	20.91	−4.872
A5	+0.078	+0.178	20.16	+1.572	B5	−0.233	−0.533	20.91	−4.872
A6	+0.013	+0.030	20.16	+0.262	B6	−0.231	−0.529	20.90	−4.828
A7	−0.037	−0.085	20.83	−0.771	B7	−0.229	−0.524	20.90	−4.786
A8	−0.069	−0.158	20.83	−1.437	B8	−0.224	−0.513	20.90	−4.682
A9	−0.092	−0.211	20.84	−1.917	B9	−0.220	−0.503	20.90	−4.598
A10	−0.125	−0.286	20.86	−2.608	B10	−0.203	−0.465	20.90	−4.243
Totals...	+0.712	.....	.....	+14.011	Totals...	−2.154	.....	.....	−45.021

Equivalent scale = (14.011 + 45.021) ÷ (0.712 + 2.154) = 20.60.

TABLE X.

Spring B.—Calculation of Equivalent Scale Corrected for all Variation in the Hot Scale of the Spring.

AVERAGE CARD, HEAD END OF LOW-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	+.325	.759	16.08	+5.226	B1	−.0 8	.171	16.81	−1.227
A2	+.282	.659	16.12	+4.546	B2	−.263	.614	16.98	−4.466
A3	+.245	.572	16.19	+3.967	B3	−.285	.666	17.00	−4.845
A4	+.201	.470	16.30	+3.276	B4	−.291	.680	17.01	−4.950
A5	+.132	.308	16.49	+2.177	B5	−.291	.680	17.01	−4.950
A6	+.048	.112	16.50	+0.792	B6	−.290	.678	17.01	−4.933
A7	−.015	.035	16.81	−0.252	B7	−.287	.671	17.00	−4.879
A8	−.060	.140	16.81	−1.009	B8	−.284	.664	17.00	−4.828
A9	−.098	.229	16.81	−1 647	B9	−.280	.654	16.99	−4.757
A10	−.135	.315	16.84	−2.273	B10	−.257	.600	16.97	−4.361
Totals....	+.925	.....	.....	+14.803	Totals ...	−2.601	.....	.....	−44.196

Equivalent scale = (14.803 + 44.196) ÷ (.925 + 2.601) = 16.73.

TABLE X.—*Continued.*

AVERAGE CARD, CRANK END OF LOW-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	+.345	.797	16.07	+5.544	B1	−.146	−.337	16.85	−2.460
A2	+.298	.688	16.11	+4.801	B2	−.284	−.656	16.99	−4.825
A3	+.255	.589	16.16	+4.121	B3	−.298	−.688	17.01	−5.069
A4	+.187	.432	16.35	+3.057	B4	−.298	−.688	17.01	−5.069
A5	+.082	.189	16.50	+1.353	B5	−.298	−.688	17.01	−5.069
A6	0	0	16.66	0	B6	−.298	−.688	17.01	−5.069
A7	−.063	.145	16.81	−1.059	B7	−.297	−.686	17.01	−5.052
A8	−.098	.226	16.81	−1.647	B8	−.293	−.677	17.00	−4.981
A9	−.125	.289	16.83	−2.104	B9	−.284	−.656	16.99	−4.825
A10	−.168	.388	16.88	−2.836	B10	−.267	−.617	16.98	−4.534
Totals	+.713	.....	.....	+11.230	Totals	−2.763	.....	.....	−46.953

Equivalent scale = (11.230 + 46.953) ÷ (.713 + 2.763) = 16.74.

TABLE XI.

SPRING C.—Calculations of Equivalent Scale Corrected for all Variations in the Hot Scale of the Spring. Areas measured from line 15 lbs. above the atm.

LARGEST CARD, CRANK END OF HIGH-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	0.755	1.736	60.17	45.428	B1	+0.178	0.409	59.79	+10.643
A2	0.714	1.641	60.22	42.997	B2	+0.017	0.039	59.88	+ 1.018
A3	0.670	1.540	60.27	40.381	B3	+0.015	0.034	59.88	+ 0.898
A4	0.549	1.262	60.24	33.072	B4	+0.015	0.034	59.88	+ 0.898
A5	0.382	0.878	59.97	22.909	B5	+0.014	0.032	59.88	+ 0.838
A6	0.280	0.644	59.78	16.738	B6	+0.009	0.021	59.88	+ 0.539
A7	0.205	0.471	59.77	12.253	B7	+0.008	0.018	59.88	+ 0.479
A8	0.148	0.340	59.82	8.853	B8	+0.002	0.005	59.88	+ 0.120
A9	0.105	0.241	59.85	6.284	B9	−0.004	0.009	59.88	− 0.240
A10	0.075	0.172	59.88	4.491	B10	−0.009	0.021	59.88	− 0.539
Totals	3.883	.....	.....	233.406	Totals	+0.245	.....	.....	+14.654

Equivalent scale = (233.406 − 14.654) ÷ (3.883 − 0.245) = 60.130.

SMALLEST CARD, CRANK END OF HIGH-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Correspond'g Scale.	Area × Scale.
A1	0.752	1.729	60.18	45.255	B1	+0.175	0.402	59.80	+10.465
A2	0.710	1.632	60.22	42.756	B2	+0.017	0.039	59.88	+ 1.018
A3	0.665	1.529	60.27	40.080	B3	+0.015	0.034	59.88	+ 0.898
A4	0.504	1.159	60.17	30.326	B4	+0.015	0.034	59.88	+ 0.898
A5	0.348	0.800	59.90	20.845	B5	+0.014	0.032	59.88	+ 0.838
A6	0.245	0.563	59.74	14.636	B6	+0.009	0.021	59.88	+ 0.539
A7	0.180	0.414	59.79	10.762	B7	+0.005	0.011	59.88	+ 0.299
A8	0.128	0.294	59.84	7.660	B8	+0.001	0.002	59.88	+ 0.060
A9	0.087	0.200	59.87	5.209	B9	−0.002	0.005	59.88	− 0.120
A10	0.057	0.131	59.88	3.413	B10	−0.003	0.007	59.88	− 0.180
Totals	3.676	.....	.....	220.942	Totals	+0.246	.....	.....	+14.715

Equivalent scale = (220.942 − 14.715) ÷ (3.676 − 0.246) = 60.124.

Average equivalent scale = (60.130 + 60.124) ÷ 2 = 60.13.

TABLE XII.

SPRING D.—Calculation of Equivalent Scale Corrected for all Variations in the Hot Scale of the Spring. Areas measured from line 15 lbs. above the atm.

AVERAGE CARD, HEAD END OF HIGH-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Corre-spond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Corre-spond'g Scale.	Area × Scale.
A1.....	.698	1.631	59.64	41.629	B1.....	+.173	.404	62.25	+10.769
A2.....	.680	1.589	59.72	40.610	B2.....	+.006	.014	64.10	+ .385
A3.....	.652	1.523	59.85	39.022	B3.....	+ 003	.007	64.10	+ .192
A4.....	.525	1.227	60.17	31.589	B4.....	+.003	.007	64.10	+ .192
A5.....	.370	.864	60.51	22.389	B5.....	+.002	.005	64.10	+ .128
A6.....	.249	.582	60.97	15.182	B6.....	−.002	.005	64.10	− .128
A7..	.168	.393	62.32	10.470	B7.....	−.006	.014	64.10	− .385
A8.....	.117	.273	63.22	7.397	B8.....	−.012	.028	64.10	− .769
A9.	.078	.182	63.91	4.985	B9.....	−.015	.034	64.10	− .962
A10.....	.047	.110	64.10	3.013	B10.....	−.018	.042	64.10	− 1.154
Totals...	3.584	.....	.....	216.286	Totals...	+.134	....	.....	+ 8.268

Equivalent scale = (216.286 − 8.268) ÷ (3.854 − .134) = 60.29.

TABLE XIII.

SPRING E.—Calculations of Equivalent Scale Corrected for all Variations in the Hot Scale of the Spring. Areas measured from line 15 lbs. above the atm.

LARGEST CARD, HEAD END OF HIGH-PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height	Corre-spond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Corre-spond'g Scale.	Area × Scale.
A1	.738	1.724	57.89	42.723	B1	+.181	.423	61.78	+11.182
A2	.703	1.642	57.95	40.739	B2	+.005	.012	63.69	+ .318
A3	.672	1.570	58.01	38.983	B3	+.004	.009	63.69	+ .255
A4	.567	1.325	58.55	33.198	B4	+.004	.009	63.69	+ .255
A5	.398	.930	59.77	23.788	B5	+.003	.007	63.69	+ .191
A6	.273	.650	60.47	16.811	B6	0	0	63.69	0
A7	.202	.472	61.42	12.407	B7	−.002	.005	63.69	− .127
A8	.151	.353	62.28	9.404	B8	−.004	.009	63.69	− .255
A9	.098	.227	63.18	6.192	B9	−.005	.012	63.69	− .318
A10	.071	.166	63.67	4.521	B10	−.018	.042	63.69	− 1.146
Totals...	3.878	.....	. .....	228.766	Totals..	+.168	.....	.....	+10.355

Equivalent scale = (228.766−10.355) ÷ (3.878−0.168) = 58.87.

SMALLEST CARD, HEAD END OF HIGH PRESSURE CYLINDER.

Division of Card.	Area.	Mean Height.	Corre-spond'g Scale.	Area × Scale.	Division of Card.	Area.	Mean Height.	Corre-spond'g Scale.	Area × Scale.
A1	.710	1.663	57.94	41.137	B1	+.169	.396	61.97	+10.473
A2	.681	1.595	57.99	39.491	B2	+.005	.012	63.69	+ .318
A3	.662	1.550	58.03	38.416	B3	+.004	.009	63.69	+ .255
A4	.531	1.244	58.81	31.228	B4	+.004	.009	63.69	+ .255
A5	.349	.817	60.05	20.957	B5	+.004	.009	63.69	+ .255
A6	.238	.557	60.81	14.473	B6	0	0	63.69	0
A7	.172	.403	61.92	10.650	B7	−.004	.009	63.69	− .255
A8	.115	.269	62.89	7.232	B8	−.005	.012	63.69	− .318
A9	.075	.176	63.54	4.766	B9	−.007	.016	63.69	− .446
A10	.047	.110	63.69	2.993	B10	−.018	.042	63.69	− 1.146
Totals...	3.580	.....	.....	211.343	Totals...	+.152	.....	.....	+ 9.391

Equivalent scale = (211.343−9.391) ÷ 3.580−0.152) = 58.91.

Average equivalent scale = (58.87 + 58.91) ÷ 2 = 58.89.

DISCUSSION.

Mr. Louis S. Wright.—On the first page of this paper it is stated "that the feed-water was measured by a carefully calibrated meter placed between the feed-pumps and boilers." A case has recently come to my notice of a meter which was calibrated first from a carefully measured tank containing about 12,000 gallons, where the water of said tank was allowed to run through the meter; again calibrated in service between the pump and boiler by another and smaller measured tank. Results in the two cases were widely different; and later, from another test similar to the last mentioned, another result was obtained, so that the meter was found to be entirely unreliable. From this, and from other cases that I have heard of, I feel that a meter as a means of measuring water pumped to a boiler is not a reliable and trustworthy way of recording the amount used.

Mr. George I. Rockwood.—I was lately engaged to make a test of a steam plant owned by a gentleman who thought it quite uneconomical to follow out my dictation in regard to providing the necessary scales and barrels. He thought that as his factory water-meter was adjacent to the boilers he could most quickly disconnect the various pipes leading elsewhere than to the boilers, and use the meter instead of erecting the expensive weighing outfit. He observed that he thought it useless to go to this "refinement" of weighing the water when he had a good meter right on the premises. As he was willing to pay for my time, I felt disposed to let him learn, after putting in my objections, and we commenced the test, gathering the data, taking diagrams, etc., for an hour or so, when the man who was reading the meter came to say that he guessed he didn't know how to read it, or else it was not metering any water. Sure enough, we found that the small constant stream of water used in the boilers was not enough to make this meter budge, although when there came any sudden and large demand for water the buckets moved fast enough, and made quite a clatter.

This circumstance does not prove that all meters are incapable of correctly measuring the flow of water through them; but, taken in connection with the paper, what it shows is, that there are meters and meters, and the lesson is, that we must know who places the confidence in the particular meter under notice before judging of its liability to error.

Having heard something of the history of the meter used in the test of this 1600 H.P. engine, and knowing the men who saw fit to employ it in so important a place, I feel no hesitation in accepting the feed-water measurements given as being very near to the truth. The steam consumption found—13.5 lbs. per I.H.P. per hour—is, of course, very low; and the coal consumption is still more remarkable, even for a full jacketed compound engine, although this engine had no jackets whatever. In my judgment the fine results are due to four circumstances, viz.: tight valves and pistons; rather a high steam pressure; superheated steam from the boilers; all in an engine of unusual size.

Whatever our theories may be as to the precise effects of superheated steam in reducing cylinder condensation, it is evident that the condensation in both cylinders, but more especially that in the first, is much less than is customary, and we cannot but believe that it was the superheat of the steam, coupled with the larger size of the engine, which so reduced the cylinder condensation.

I think that the fact that this engine has shown such good results need not interfere with our general faith in the triple-expansion engine. I am very sure it does not interfere with my faith in the two-cylinder compound designed with large superheating intermediate receiver and great disparity in sizes of cylinders, simply because I have obtained results 7% better with steam quite wet instead of superheated, as in this case, and of a pressure of 140 lbs.

Mr. Chas. E. Emery.—I wish to thank the authors of this paper, and particularly Professor Jacobus, the author of the former paper, DLXXVIII., for the very excellent information in relation to the errors of the steam-engine indicator. I did not have time to examine the paper last named until after the meeting, but it explains the probable reason why, in my early experience with indicators, I did not have the difficulties stated in connection with modern high-speed indicators. The advance in mechanical workmanship permits fits that were reserved for gauges a few years ago, and prohibited in every-day work, so that now pistons which are free to move are still too tight for the duty they have to perform. I always had my indicator pistons quite leaky, and drilled extra holes above the same to carry off leakage; and the use of springs mechanically fastened at the ends, and then soldered so that there was no slack motion in the fastenings, gave good results throughout the whole range of the spring. There is

a reason in addition to those given in the paper why calibration should be made at one-half the maximum pressure. The average temperature of an indicator piston, cylinder, and spring probably does not reach that due to the maximum steam pressure.

Recurring to the paper in question, I wish to commend the authors for a very valuable account of interesting tests which serve to increase our stock of knowledge on a very important subject.

Mr. F. H. Ball.—This paper records the remarkable performance of a compound engine, and is therefore a valuable contribution to the published transactions of our society.

Whether the facts here presented shall result in advancing the science of steam-engineering or not, depends on our ability to correctly diagnose the case and to point out the conditions which contribute most to this phenomenal showing.

It is to be regretted that the gentlemen to whom we are indebted for this interesting paper have not seen fit to advance any theory to account for the unusual economy of this engine.

It is supposed to be the instinct of an engineer to look behind and beyond every phenomenon for its cause.

Franklin's method when seeking an answer to the query as to why a fish placed in a bucket of water did not add to the weight, was first to see if it is true.

In this case we are relieved from this first step of the investigation by reason of the evident care with which the authors of this paper have weighed the water and the fish for us, so we can proceed with the second step of the inquiry.

Referring to the record that is before us, it appears that a horse power is developed on 13½ lbs. of steam per hour, which is very close to the best recorded performance of triple expansion engines in which more than 20 expansions are generally obtained, therefore the efficiency of this engine cannot be due to the number of expansions that are realized. Neither can it be due to small clearance, as the engine does not excel in this respect. The absence of steam jackets will hardly be claimed to account for the economy, even by those who do not find any gain in the jacket.

It is true that high piston speed is obtained, but this is due to the unusual length of the cylinder, and therefore would warrant the expectation of greater cylinder condensation rather than less, because of the longer time in which the steam is in contact with the cylinder walls. From a study of the diagrams, it does not

appear that the steam distribution was remarkably perfect; in fact, the loss between the diagrams of Fig. 254 is more than is usually found in engines of this class. It will be noticed, also, that compression in both cylinders is carried to initial pressure, which, in view of the facts regarding compression that are presented by Professor Jacobus in his paper on that subject, cannot be considered as the most economical compression. So far, then, no good reason appears for the economy of this engine. Referring to the diagrams of Figs. 250 and 251 from the high pressure cylinder it will be noticed that expansion is not carried to the line of the return pressure, but a considerable terminal "drop" occurs, and this fact is suggested as the probable reason for the efficiency of this engine. In other words, the high pressure cylinder is developing power more economically than where expansion is carried too far in this cylinder.

To corroborate this idea, it will be noticed that in the tests of compound engines reported to this Society by Mr. Rockwood, showing high efficiency, the cylinder ratios are large, and a considerable terminal drop occurs. He substitutes one cylinder working under conditions favorable for high efficiency, for two cylinders in which expansion is carried too far, and thus obtains with the compound engine practically the same economy as the triple engine develops. I cannot, however, accept his conclusion as to the relative efficiency of the two engines, because it has not been shown that the same judicious selection of cylinders for the triple engine would not improve its performance and make a new record for it, and I am of the opinion that it would.

*Professor Jacobus.**—Mr. Wright has stated that in his experience he has found the meter to be an unreliable means of measuring the water used by the engine. Of course there are meters which do not work properly; but the fact that a meter sometimes gives erroneous results should not be brought up as an argument that all meters are unreliable. The way we calibrate a meter is this: the meter is always placed on the pressure side of the pump, and it must be put in such a way that it is subjected to no severe water-hammer. The feed-pump should run smoothly. The pistons of the meter should hit the heads at the end of each stroke, and no air should be allowed to collect in the meter. We calibrate a meter a number of times under actual conditions which

*Author's Closure, under the Rules.

exist during a test. The water is drawn out through a nozzle, the pressure and rate of flow being preserved exactly the same as existed throughout the test.

Then we take an average of the calibrations.

The figure obtained for the engine economy, or $13\frac{1}{2}$ lbs., per hour per horse power, is certainly a low one, so that if any mistake were made in the meter it would appear from this figure that the meter should have registered more water than the amount indicated by its readings.

The rate of evaporation determined by the same meter is $10\frac{1}{3}$ lbs. per pound of coal under actual conditions. If the meter registered too low it would follow that the $10\frac{1}{3}$ ought to be greater, but as this rate of evaporation is very high, and all the heat in the coal, as determined by calorimetric test, was accounted for, except a small amount for radiation, which fell in line with other experiments, it appears that the meter could not have registered too small an amount. In other words, the result for steam per hour per horse power is so low that if there is an error in the meter it probably registered too small an amount of water, whereas an analysis of the results obtained at the boilers shows that this was not the case.

Here is another check that was made on the meter: there were eight boilers. One was disconnected, and the water fed to it was weighed. The rate of evaporation of the single boiler was precisely the same as obtained by weighing the water as that obtained in the eight when the water was measured by the meter.

Still another reason why we consider this meter correct is this: we have used it in a number of previous tests, and it has been used in one test since it was in use at the Bristol Mill, and we know it records at a constant rate, because it always gives practically the same rate when standardized under a given set of conditions.

Another thing that should be noticed is, that the meter was balanced up every half hour during the test, and the half hour rates were always uniform, which shows that it could not have varied in its readings. The way that some people have obtained erroneous results with meters is to place the meter on the suction side of the pump, and in this case air often lodges in the meter and cannot be removed. I know of one case where the meter was used to measure the water delivered by a pump which drew water

from a hot-well. The water in the hot-well would sometimes fall below the level of the suction pipe so that air was drawn directly into the meter. In this case very abnormal results were obtained.

A meter should be as carefully handled and as accurately standardized as a steam engine indicator, and a person who is capable of using one of these instruments with accuracy should be capable of doing the same with the other.

Mr. Ball has suggested that the terminal drop at the end of expansion in the high-pressure cylinder may account for the high economy of the engine. There are as yet no data to support this view. An examination of the results of the tests shows, as Mr. Rockwood has stated, that the high economy is probably due to the unusual size of the engine, together with the small amount of cylinder condensation, which was probably diminished below that usually found in Corliss engines, by the effect of the high speed combined with the superheated steam furnished by the boilers. The Willans engine presents a similar case, the high economy obtained being ascribable only to the low per cent. of cylinder condensation.

On comparing the results of the test in question with others on compound engines, it may be seen that the economy is less, by a small amount, than that obtained with jacketed engines. For example, in a test made by Mr. Geo. H. Barrus on a Corliss cross-compound engine located at the plant of the Dwight Manufacturing Co., Chicopee, Mass., 13.26 lbs. of feed-water was consumed per hour per horse power.* This engine developed 1,018 horse power. The piston speed was about 700 feet per minute; boiler pressure 119.8 lbs. The steam was superheated 44.5° Fahr. The high-pressure cylinder was jacketed with high-pressure steam, all the steam passing through the jacket, and finally entering the steam chest together with the water of condensation. The low-pressure cylinder was jacketed in a similar way with the exhaust steam from the high-pressure cylinder. All the water of condensation of the jackets passed through the cylinders. The ratio of the cylinders was about 3 to 1.

Another case, not so nearly parallel on account of the higher boiler pressure and smaller size of the engine, is that of one of Mr. Rockwood's engines, in which the steam consumption was

* The *Engineering Record*, Nov. 5, and Nov. 12, 1892.

12.85 lbs. per hour per horse-power, when 194 horse-power was developed with a boiler pressure of 143 lbs.*

Comparing our test with that by Mr. Barrus, we find a difference in economy of about two per cent. in favor of the jacketed engine, and this difference might be greater if the engines were of the same dimensions.

* Transactions, Vol. xiv., page 465.

DXCII.*

THE SATURATION CURVE AS A REFERENCE LINE FOR INDICATOR DIAGRAMS.

BY R. C. CARPENTER, ITHACA, NEW YORK.
(Member of the Society.)

THE following paper is intended to describe a graphical system which the writer has employed for obtaining more complete information regarding the heat interchanges which occur in the steam engine than is to be had by the usual method of reducing the data of an engine test. During the past year, the students in laboratory practice of steam engineering at Sibley College have all been required to make a graphical thermal analysis, based on the use of the saturation curve as explained in the following article. The article is intended simply to give information showing the use of the method, and some examples are given which show the character of the results which can readily be obtained. No attempt is made to show by these examples any results which have a general application, and it is not expected that any conclusions except such as relate to the method itself will be drawn from the tests presented. In apology for the elementary way in which the very simple system is described, I may say that the article is intended principally for students in technical schools.

This system cannot be considered as new in its general features, as I find that it has been used in a somewhat similar way both by Professor Unwin and Professor Cotterell, but it has been helpful in many ways as a rapid and graphical method of determining the quality and thermal condition of the steam during expansion, and in some respects the method presents novel features. It seems to have been used very little, possibly not at all, by American engineers. Thinking that a method which we have found useful might prove of interest, especially to technical students, I have ventured to describe it.

The method is, itself, quite simple and can be readily explained

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

by reference to the diagrams in Fig. 255. In these diagrams *aormn* is the actual diagram obtained from the engine, *KR* represents the atmospheric line, *bb* the clearance line, and *K'R'* represents the line of no pressure or vacuum line, *ob* represents the volume of clearance, and *ON* represents the volume corresponding to piston displacement. From the end of the compression curve an hyperbola is produced backward and upward, a_5a, to the line *bc*, which latter corresponds to the boiler pressure of steam. The volume of steam filling the clearance space corresponds with very little error to the distance *ab* on the same

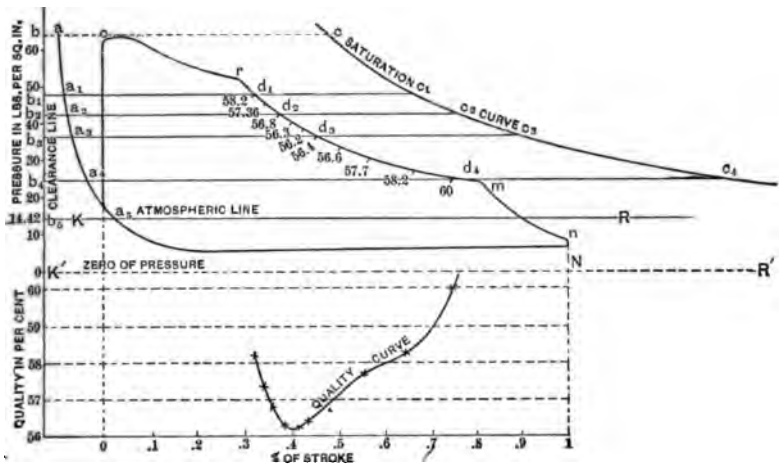

FIG. 255.

scale as the volume of piston displacement. To use this method, the weight of steam used per stroke is required. This is to be found by an ordinary efficiency test of the engine, either by condensing the steam, or from a boiler test. From this data we can compute the weight of steam per stroke used by the engine. By consulting steam tables the volume corresponding to the weight for the given pressure can be ascertained. This volume is then laid off to the same scale as the diagram in a horizontal direction from the point which corresponds to the volume of steam filling the clearance at boiler pressure. In the diagram under consideration, this volume corresponds to the distance *ac*, which is laid off from the point *a* for the reason stated. The values given in the steam table for different pressures, multiplied by factors for weight of steam and for the scale used, give

abscissae of the other points in the saturation curve. In the test represented by the diagrams in Fig. 255, we have for the saturation curve the line $cc_1 c_2 c_3 c_4$. The abscissae of this curve for any given pressure ordinate shows the total volume that the same weight of dry and saturated steam would have had, to the same scale of volume as the piston displacement. If now a horizontal line be drawn extending from the clearance line to the saturation curve, the ratio borne by the distance from the clearance line to the expansion curve to the whole line is the quality of the steam.* Thus through the point d_1, Fig. 255, draw the horizontal line, $b_1 c_1$, then the ratio of $b_1 d_1$ to $b_1 c_1$ is the quality at the point d_1; in the same manner $b_2 d_2 \div b_2 c_2$ is the quality at d_2, etc. The variation in quality from point to point as obtained in that manner is shown for different positions in the stroke by the curve below the diagram, from which it is seen that for the given case the quality during the early periods of expansion rapidly diminished, then gradually increased, so that just before the period of exhaust opening the quality was higher than that at the beginning. From a study of the conditions the same method could be applied during the period of compression, the only difficulty being this, that the quality of steam at any point in the cylinder during compression is not positively known, and an assumption must be made which is perhaps no better than to assume that during compression the curve is hyperbolic. The weight of steam imprisoned in the clearance spaces is always a very small percentage of the total, hence any error which is made in assuming hyperbolic compression cannot affect the results seriously, in fact, it becomes of very little moment.

There is very much in this graphic analysis which brings out thermal properties in a way better calculated to make an impression upon the mind than the analytic and more difficult methods of Hirn's analysis. We have employed this method in the Sibley College tests as supplementary to the analytical method devised by Hirn. So far as the quality shown during expansion, the two methods have always given substantially the same results, the difference, in fact, depending simply upon the accuracy with which either the computation or the drawing was made. In the test for Hirn's analysis, the total heat discharged from the engine is

* There is a slight error in this assumption, due to the volume of the entrained water. The error is in general less than can be measured. A table for correction is given at the end of the article.

always measured or computed so that the quality of the exhaust steam can be determined. The same data could be used in conjunction with the graphical method described, in which case the heat interchanges could be very readily and quickly computed, using the diagrams to obtain the quality at any given point. Thus, if from an ordinary test the weight of steam at any point is given and the quality is shown by the graphical method which has been explained, the total heat is computed from a simple formula in the following manner:

Thus, let $x =$ the quality as determined from the diagram, ρ the internal latent heat, q the sensible heat, for the given pressures, the latter being obtained from a steam table. Then the total heat in one pound of steam will be $x\rho + q$, which can be computed from the data at hand. If r is the total latent heat, the total heat available after condensation would be $xr + q$.

While the graphical method will not replace the more tedious and exact analytical one for all purposes, yet it is believed that it will give all the practical results which can be obtained from the analytical method with much less work and, in general, in a manner which is more readily understood by the student in technical schools. This method, I think, will, on the whole, prove of some value to practical engineers. This, I believe, has not been the case with the analytical method of determining the heat losses.

As another illustration of the method of using saturation curves, reference is made to Figs. 256 and 257, in which case we have the same engine working with the same brake load and practically the same I. H. P. The load was very light; in the one case the engine was worked with very little compression, and in the other (see Fig. 257) with a great deal of compression. The discussion here is merely to show the method of obtaining the results by a graphical process, and no reference is made to the relative economy in the two trials, for the reason that it will probably form the subject of a paper before the Society by one of my colleagues in Sibley College. In these figures "A," Fig. 256, is an exact copy of the diagram, on reduced scale, as obtained from the indicator. In Fig. 256, diagram "B," the line AB represents the expansion curve, KL, the saturation curve being drawn to the same scale of volumes. The quality at various points on the expansion line is denoted by figures and also represented by the curve, diagram "C," at the top of the figure. In Fig. 257

is shown, to reduced scale, the diagram, actual and enlarged; the saturation curve, corresponding to the weight of steam impris-

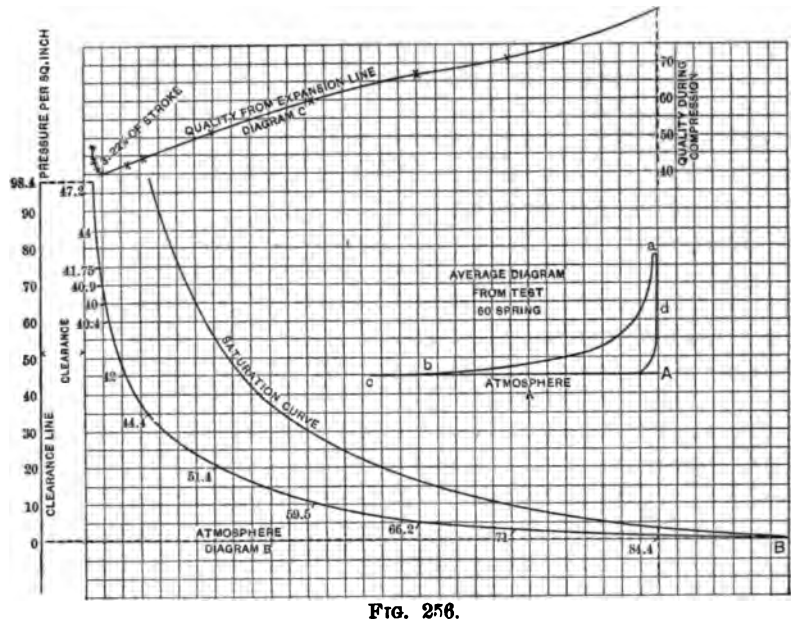

FIG. 256.

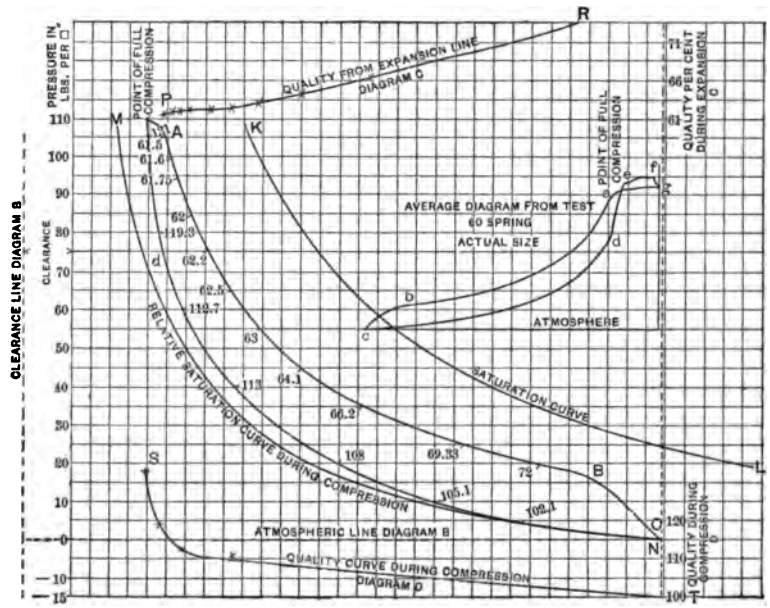

FIG. 257.

oned in the clearance spaces, is drawn on the supposition that this steam is dry at the beginning of compression. The quality at different points in the compression curve and at different points in the expansion line is denoted by figures on the enlarged diagram. The quality is also shown graphically by curves at the upper and lower portions of the figure. While no discussion of the results can be made here, it is interesting to note the improved character of the quality of steam produced by the late compression. It should not, however, be concluded that the one case is for this reason much superior in economy to the other, as I believe that certain compensating disadvantages were experienced which made the economical results practically the same in both cases.

The application of this method to a compound or triple expansion engine is quite similar to that described for the simple engine. In general, it will be found very much better to have the ordinates representing pressures drawn to as large a scale as possible, since in this case both the expansion curve and the saturation curve become more nearly vertical, and the intersection of horizontal lines becomes better defined, and is much more accurate. It has been our custom to treat each diagram from a compound or triple expansion engine, as already explained for the simple engine, and to obtain the positions of the saturation curve before the combined diagram, showing all the cards to the same scale, was constructed.

In making the combined diagram, we have tried various plans in order to have a continuous curve of reference to which it might be referred. To do this it is necessary to arrange the diagrams with reference to different clearance lines. Even then, because of the different weights of steam caught in the clearance spaces, the saturation curve could not be continuous, or, if adjusted so as to be continuous, it would be of different curvature for the part corresponding to each cylinder. This later objection will not hold when the hyperbola is used as a reference line. The result of various experiments in combining diagrams has been to satisfy me that the method of drawing the diagrams to the same pressure and volume scale is on the whole as good as any that has been devised. This method, however, gives us a broken reference curve, as will be seen by referring to Fig. 258, which shows the combined diagrams from the Sibley College experimental engine. On the whole, it seems

to be the only method of representing the combined diagrams which is general in its nature. It is true that very much smoother reference curves can be obtained for the combined diagram, which, for some cases, may more fairly represent the condition of the engine, by arranging the diagrams in such a manner that a continuous reference curve, either hyperbolic or

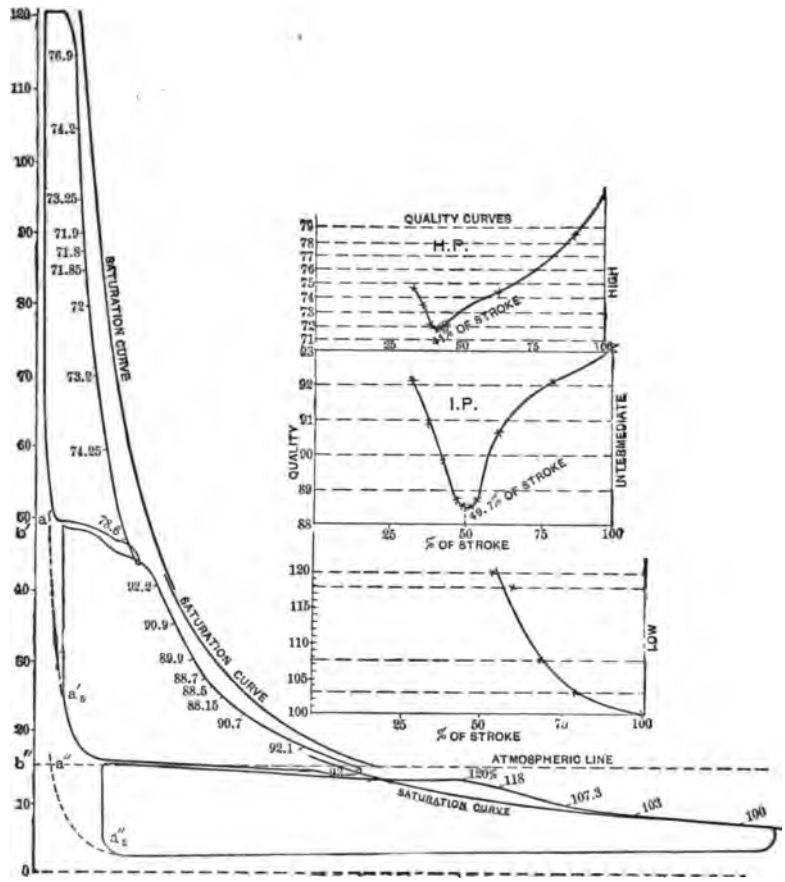

FIG. 258.

in the form of the saturation line, is obtained. But it is believed that, on the whole, no injustice is done by the distortions which sometimes occur in the application of the method of applying uniform scale of pressure and volume.

In Fig. 258 are shown the combined diagrams from the high,

intermediate, and low pressure cylinders of the triple expansion engine at Sibley College. The saturation curve drawn as already explained for each diagram separately is given, and the corresponding quality is marked on the expansion curves for various points in each cylinder. The curves of quality are shown for the high and intermediate cylinders to the right of the diagram. The engine, in this case, was steam jacketed with high pressure steam on all the cylinders and in both receivers. The effect of the jacket steam on the quality is very marked, as is shown by reference to the expansion curve of the low pressure cylinder. It will be noted in this case that the figures indicate in some stages a quality exceeding one hundred. The numbers on the low-pressure diagram should be considered as representing the ratio of volume to that in the same weight of dry and saturated steam, and not as representing the per cent. of dry steam or of degrees of superheat. This can be taken merely as representing a superheated condition without a specific statement of the degree of superheat. It is not certain that the degree of superheat can be determined by increase of volume, since superheated steam does not follow the law of a perfect gas until it is superheated to a great amount. If superheated steam expanded according to the same law as perfect gases, $pv \div T$ would be a constant, and for given values of p and v, T could be computed. See Note II.

The peculiar form of the quality curves shown in the figures as pertaining to the high and intermediate cylinders seems very generally true for the usual conditions of the steam engine. In a great many trials that we have considered, the curves have the form shown in each case.

The point of contraflexure in these curves usually occurs some little distance after the cut-off, and indicates the position in the cylinder where the quality has become constant, and condensation has ceased and re-evaporation has begun. The peculiar curvature occurring after re-evaporation begins is no doubt largely due to the rapid motion of the piston when near the centre of the stroke and to its very slow motion near the end.

These few remarks serve to show in a general way the character of the analysis and the results which may be obtained by using a saturated curve as explained. I think it will be conceded that, as a method, it is certainly worth careful considera-

tion. The information regarding the interchanges of heat cannot be obtained so readily by any other method that is known to the writer, while I believe it will be found to compare well with any for completeness and accuracy of information.

Applications of the methods which have been here described have already been presented to the Society in the discussion of the tests of the Milwaukee engine by Dr. Thurston, read at the fall meeting, and, also, a paper at the present meeting on the subject of "The Steam Jacket," also by Dr. Thurston.

In presenting the drawings for this paper the writer is under obligation to Mr. Thomas Hall, a graduate student in Sibley College.

NOTE I.—*Error in determining quality of the steam.* The quality of the steam can be defined as the percentage by weight of steam in a mixture of steam and water. Denote the quality by x, the percentage of moisture by weight, $100 - x$, the volume in cubic feet of one pound of dry and saturated steam by v, volume of one pound of water by c, the volume of one pound of the mixture by w.

Then will we have

$$xv + (100 - x)\, c = w,$$

from which

$$x = \frac{w - 100c}{v - c} = \frac{w}{v} - \left(100 - \frac{w}{v}\right) c.$$

In the above expression w corresponds to the actual volume shown by the diagram for any given pressure, v that of the corresponding volume measured to the saturation curve; the quotient $w \div v$ being the quantity which has been taken as the quality in the preceding discussion. From the above formula it is evident that the true quality will be less than the quantity so obtained by an amount the value of which is expressed in the second term of the second member. This last quantity is always very small. The value of c is nearly constant, and may be taken without sensible error as 0.016. The value of this correction does not equal 1% until the quality has become less than 40%, or until the amount of moisture exceeds 60%. The following table gives the percentage of correction for different conditions with respect to volumes of the steam.

TABLE FOR CORRECTING VALUES OF QUALITY AS OBTAINED FROM RATIO OF VOLUMES AS EXPLAINED IN PAPER.

A Percentage obtained by ratio of volumes $v' \div v$.	B Correction to be subtracted. Percentage.	C True value of the *quality*. Per cent.	A Percentage obtained by ratio of volumes $v' \div v$.	B Correction to be subtracted. Percentage.	C True value of the *quality*. Per cent.
100	0.000	100.00	70	0.48	69.52
98	0.032	97.968	65	0.56	64.44
96	0.064	95.936	60	0.64	59.36
95	0.080	94.92	55	0.72	54.28
94	0.096	93.904	50	0.80	49.20
92	0.128	91.872	45	0.88	44.12
90	0.16	89.84	40	0 96	39.04
85	0.24	84.76	35	1.04	33.96
80	0.32	79.68	30	1.12	28.88
75	0.40	74.60	25	1.20	23.80

NOTE II.—*Application to Superheated Steam.* If the volume of steam for a given pressure is greater than that obtained by measuring to the saturation curve, the steam is superheated. It is somewhat doubtful if the degree of superheat can be determined by the increase in volume. If we could consider steam when superheated as a perfect gas, the product of pressure and volume divided by absolute temperature would be constant; farther, for any given pressure the volume would vary directly as the temperature.

I have been unable to find much information relating to the properties of superheated steam. It seems, however, generally admitted that until the degree of superheat becomes considerable, the temperature does not increase at the same rate as the volume. After the steam is superheated considerably, it behaves like a perfect gas, of which the equation is $pv = 85.5\,T$ ft.-lbs., in which p is the pressure in pounds per square foot, v the volume in cubic feet, and T the absolute temperature. The range of temperature required for the steam to pass from the saturated to the gaseous condition for any given pressure is given by Hirn as 16° Fahr., and by Siemens as 18° Fahr.*

The following diagram, Fig. 259, was constructed by computing ratio of $\frac{pv}{T}$ from the values given in the steam table in

* See Thurston's *Steam Engine*, Vol. I. p. 771; *Théorie Mécanique de la Chaleur*, Ency. Britannica, Vol. XXII. p. 485.

Wood's *Thermodynamics*, and shows the relation between p, v, and T for different pressures with saturated steam. The straight line represents the equation as given above for superheated steam. It is to be noticed that the constant for saturated steam varies from 85.5 at 1 lb. pressure to 78.5 at 200 lbs. pressure.

DIAGRAM SHOWING RELATION OF PRESSURE, VOLUME AND ABSOLUTE TEMPERATURE OF SATURATED AND SUPERHEATED STEAM.

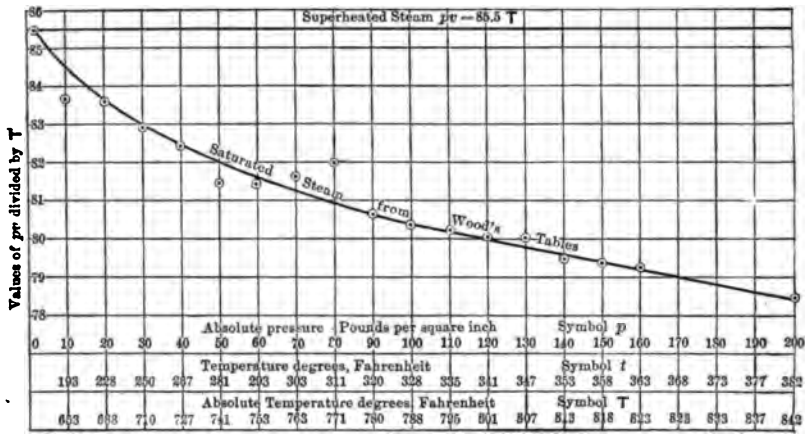

FIG. 259.

An application of these equations to the diagram of the low-pressure cylinder in Fig. 258 would indicate that the steam entering the low-pressure cylinder was 340° in temperature. As the jacket steam was only 350°, this amount seems unreasonably high.

DXCIII.*

RESULTS OF EXPERIMENTS WITH A FIFTY HORSE-POWER SINGLE NON-CONDENSING BALL AND WOOD ENGINE TO DETERMINE THE INFLUENCE OF COMPRESSION ON THE WATER CONSUMPTION.

BY D. S. JACOBUS, HOBOKEN, N. J.
(Member of the Society.)

In a paper presented to this Society at the recent Chicago meeting by Mr. F. H. Ball,† a theory was advanced in regard to the probable effect of compression on the economy of the steam engine, and a law was suggested for determining the most economical compression curve. Since that meeting, an opportunity has been afforded, at the Stevens Institute of Technology, for investigating this subject by experiment, and the following pages are a record of the facts thus obtained.

The engine used for this purpose, which will be more fully described further on, was provided with two valves for alternate use, one of them being a special valve arranged for obtaining full compression to initial pressure, as shown in Fig. 260. Owing to the rather large clearance of this engine, it was found impracticable to obtain the compression curve sought, if any later cut-off was used than the one shown, giving only about 20 lbs. M.E.P., and thus limiting the M.E.P. of Fig. 262 to about 30 lbs., which is considerably below the rated capacity of the engine at which it is supposed to give its highest economy. It was also found necessary to reduce the pressure to about 71 lbs. to obtain the desired compression, and the temporary foundation made it necessary to run the engine about 50 revolutions per minute below its normal speed; so that, because of this reduced speed, the low boiler pressure, and the small load, the consumption of steam per indicated horse-power per hour was considerably

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† *Transactions American Society of Mechanical Engineers*, Volume XIV. p. 1067, No. 545.

above what might have been expected from an engine of this class under more favorable conditions. These unfavorable conditions, however, did not present any obstacles to the investigation which was to be made regarding the relative economy of varying amounts of compression.

The experiments prove that for either equal amounts of work produced, or for equal points of cut-off, the cushion steam in an engine should not in general be compressed as high as the initial pressure in order to obtain the best economy, but to some lower pressure, thus verifying conclusions arrived at by theory.

The results obtained are as follows :

With a special valve in the engine, so arranged as to compress the cushion steam to the initial pressure, and cut-off at one-quarter stroke, the steam consumption was 37.9 lbs. per hour per horse-power.

For the same work as with the special valve, and compression to two-thirds the initial pressure, the steam consumption was 36.7 lbs. per hour horse-power.

For the same cut-off as the special valve, and compression to about five-eighths the initial pressure, the steam consumption was 34.8 lbs. per hour per horse-power.

The cylinder dimensions were: bore, 10 inches; stroke, 11 inches. The steam pressure averaged about 72 lbs. above the atmosphere, and the revolutions 250 per minute.

The average indicator cards for the three sets of conditions are given in Figs. 260, 261, and 262.

The special valve released the steam at a later point in the stroke than the regular valve, so that small corrections have been made in the above figures to eliminate variations from this cause, by extending the expansion line in each case to the end of the card, as represented in Figs. 260, 261, and 262. A Barrus calorimeter was employed to determine the amount of moisture present in the steam, and the moisture was deducted to obtain the steam used. In making the tests, the work developed by the engine was absorbed by means of a Prony brake. The exhaust steam was condensed in a surface condenser, and weighed. Indicator cards were taken every five minutes, the indicator being cleaned and oiled between each three cards. Readings of the revolutions, temperatures, pressure, etc., were also made every five minutes. The water was collected in barrels, and weighed each fifteen or twenty minutes. The

weighings agreed very well among themselves. All tests were an hour or longer in duration. In order to make certain that the tests were long enough to insure the proper accuracy, the steam per hour per horse-power was calculated for the first half of five of the runs, and the maximum discrepancy between the results for the first half of the runs and the entire runs was

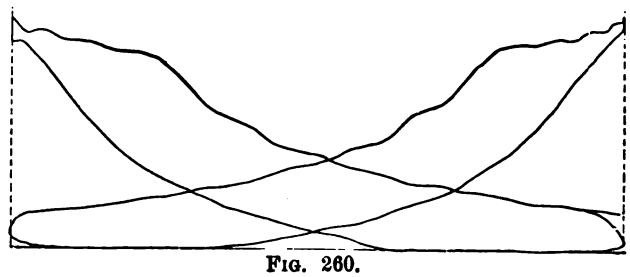

FIG. 260.

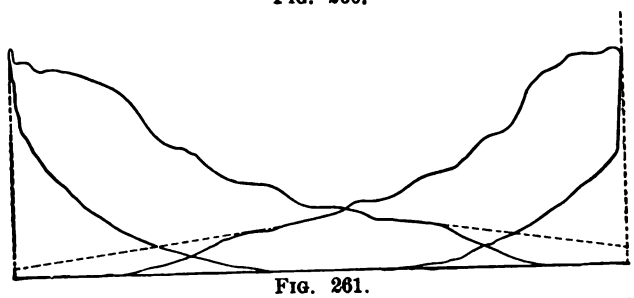

FIG. 261.

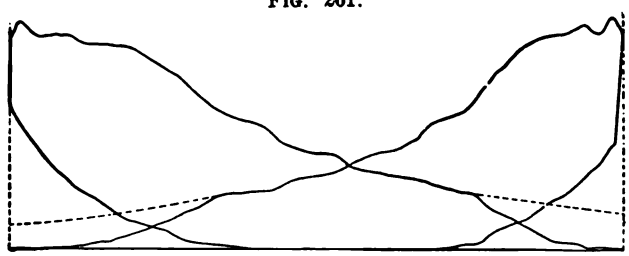

FIG. 262.

found to be one-tenth of a pound of steam per hour per horse-power, or about one-fourth of one per cent., this is less than the possible error, and thus proves that the tests were of ample length.

The results of the separate tests are given in Tables I. and II. The tests made under similar conditions are seen to agree very well among themselves, the greatest discrepancy being one per cent.

TABLE I.

FINAL RESULTS OF TESTS, CORRECTED FOR DIFFERENCES IN POINTS OF RELEASE BY EXTENDING THE EXPANSION LINES TO THE ENDS OF INDICATOR CARDS, AS IS INDICATED IN FIGURES 260, 261, AND 262.

Steam pressure at engine = 72 lbs. above atmosphere. Revolutions per minute, about 250.

CONDITIONS.	No. of Test.	Duration of test in minutes.	Average mean effective pressure in lbs. per sq. inch.	Indicated horse-power.	Steam per hour per horse-power.
Special valve so arranged as to compress the cushion steam to the initial pressure.	4 5 6	60 100 120	20.81 20.90 20.90	22.76 22.80 22.79	37.8 38.1 37.9
Average			20.87	22.78	37.9
Ordinary valve. Nearly the same mean effective pressure as for the special valve. Cushion steam compressed to about two-thirds the initial pressure.	2 7 8	80 60 60	20.37 20 20 20.02	22.65 22.00 21.62	36.8 36.8 36.6
Average			20.20	22.09	36.7
Ordinary valve. Same point of cut-off as for the special valve. Cushion steam compressed to five-eighths the initial pressure.	1 3	105 75	28.14 27.30	30.29 30.46	35.0 34.7
Average			27.72	30.38	34.8

For results in detail see Tables III., IV., and V.

TABLE II.

FINAL RESULTS NOT CORRECTED FOR DIFFERENCES IN THE POINTS OF RELEASE SO THAT THE STEAM USED PER HOUR PER HORSE-POWER GIVEN IN THIS TABLE IS THAT ACTUALLY CONSUMED BY THE ENGINE.

CONDITIONS.	No. of Test.	Duration of test in mins.	Average mean effective pressure in lbs., per square inch.	Indicated horse-power.	Steam per hour per horse-power.
Special valve so arranged as to compress the cushion steam to the initial pressure.	4	60	20 69	22.63	38.1
	5	100	20.78	22.68	38.3
	5*a*	40	20.89	22.78	38.2
	6	120	20.78	22.71	38.1
	6*a*	60	20.72	22.62	38.2
Average, not including partial runs marked *a*.....			20.75	22.67	38.2
Ordinary valve. Nearly the same mean effective pressure as for the special valve when allowance is made for differences in the points of release. Cushion steam compressed to about two-thirds the initial pressure.	2	80	18.66	20.75	40.2
	2*a*	40	18.71	20.89	40.2
	7	60	18.48	20.13	40.2
	8	60	18.31	19.77	40.0
Average, not including partial runs marked *a*.....			18.48	20.22	40.1
Ordinary valve. Same point of cut-off as for the special valve. Cushion steam compressed to five-eighths the initial pressure.	1	105	25.99	27.98	37.9
	1*a*	60	25.99	28.01	37.9
	3	75	25.23	28.15	37.5
	3*a*	45	25.08	27.99	37.6
Average, not including partial runs marked *a*.....			25.61	28.06	37.7

The averages marked *a* are for the first portions of the corresponding tests. These averages are used in calculating the performances for the first portions of the runs, which are shown not to vary over one-fourth of one per cent. from the results deduced for the entire runs. This indicates that the runs were of sufficient length. For comparative figures see Table V.

The Barrus calorimeter was attached in the main steam pipe about three feet from the steam chest. The waste steam was carried off a short distance through a three-quarter-inch pipe, which produced a slight back-pressure, so that the lower thermometer read 213° when the exit steam was saturated. The various percents of priming are given along with the average data for the tests in Table III.

Table IV. gives the average heights of the indicator cards, and Table V. the mean effective pressures and calculations of horse-power. Table VI. gives the data observed, and Table VII. the measurements of the indicator cards for one of the tests.

The indicator spring was tested over the entire range of the indicator cards, and all variations in the scale were corrected for

by employing the scale corresponding to each ordinate of the forward and back pressure lines, as explained at the last meeting of this Society in a paper on the Comparison of Indicators. Cards for each set of conditions were taken, and it was found that the differences in the equivalent scales corrected for all variation between the cards under the several sets of conditions, and between the largest and smallest cards of a set, were within the limits of practical errors. The average of the scales, or 58.2 lbs. per inch, was, therefore, used in computing the horse-power. The calculation of the equivalent scales for one of the cards is given in Table VIII.

TABLE III.

AVERAGE DATA OF TESTS AND WEIGHT OF STEAM LESS ENTRAINED WATER.

Number of Test.	Date.	Conditions.	Duration of test in mins.	Total number of revolutions by counter.	Average pressure of steam at engine in pounds per square inch above atmosphere.	Total weight of steam and entrained water in pounds.	Average temperature of exit steam at calorimeter.	Percentage of moisture in steam.	Steam per hour less entrained moisture.
1	March 10	Equal cut-off	105	26,186	70.8	1,881¼	253	1.4	1,060.2
1a	" 10	" "	60	14,976	70.8	1,076	253	1.4	1,060.9
2	" 12	Equal M.E.P.	80	20,606	72.1	1,146	227	2.9	834.6
2a	" 12	" "	40	10,342	72.1	576½	224	3.0	838.8
3	" 15	Equal cut-off	75	19,377	71.2	1,322½	276	0.1	1,056.7
3a	" 15	" "	45	11,629	71.0	793½	271	0.4	1,053.8
4	" 16	Special valve	60	15,205	71.8	870	259	1.0	861.3
5	April 3	" "	100	25,280	71.7	1,475½	244	1.8	869.4
5a	" 3	" "	60	15,148	71.8	888½	242	2.0	870.5
6	" 4	" "	120	30,383	72.2	1,741½	268	0.6	865.5
6a	" 4	" "	60	15,175	71.7	868½	268	0.6	863.0
7	" 4	Equal M.E.P.	60	15,136	72.1	824½	248	1.7	810.2
8	" 4	" "	60	15,015	71.5	796½	265	0.7	790.7

A small variation in the amount of work done will cause a greater discrepancy in the results than the difference between the economy at full and for partial compression; so that unless the greatest care be exercised in performing the experiments, the comparative results may be misleading.

TABLE IV.

INDICATOR CARD MEASUREMENTS.

Number of Test.	Mean Height of Indicator Cards in Inches.				Mean Effective Pressures in Lbs. per Square Inch. Scale = 58.2.			
	Head End.		Crank End.		Head End.		Crank End.	
	Actual.	Corrected for release.	Actual.	Corrected for release.	Actual.	Corrected for release.	Actual.	Corrected for release.
1	0.468	0.505	0.425	0.462	27.24	29.89	24.74	26.89
1a	0.468	.....	0.425	.....	27.24	.....	24.74	.....
2	0.339	0.369	0.302	0.331	19.73	21.48	17.58	19.26
2a	0.340	.....	0.303	.....	19.79	.....	17.63	.....
3	0.466	0.501	0.401	0.437	27.12	29.16	23.34	25.43
3a	0.464	.....	0.398	.....	27.00	.....	23.16	.....
4	0.370	0.373	0.341	0.342	21.53	21.71	19.85	19.90
5	0.369	0.372	0.345	0.346	21.48	21.65	20.08	20.14
5a	0.371	.....	0.347	.....	21.59	.....	20.20	.....
6	0.368	0.371	0.346	0.347	21.42	21.59	20.14	20.20
6a	0.367	.....	0.345	.....	21.36	.....	20.08	.....
7	0.332	0.362	0.303	0.332	19.32	21.07	17.63	19.32
8	0.329	0.359	0.300	0.329	19.15	20.89	17.46	19.15

TABLE V.

CALCULATIONS OF INDICATED HORSE-POWER AND STEAM PER HOUR PER HORSE-POWER.

Number of Test.	Duration in Minutes.	Revs. per Minute.	Indicated Horse-power.						Steam per hour less entrained moisture.	Steam per Hour per Horse-power.	
			Actual.			Corrected for release.					
			Head end.	Crank end.	Total.	Head end.	Crank end.	Total.		Actual.	Corrected for release.
1	105	249.4	14.82	13.16	27.98	15.99	14.30	30.29	1,060.2	37.9	35.0
1a	60	249.6	14.84	13.17	28.01	.....	.....	.....	1,060.9	37.9	....
2	80	257.6	11.09	9.66	20.75	12.07	10.58	22.65	834.6	40.2	36.8
2a	40	258.6	11.17	9.72	20.89	.....	.....	.....	838.8	40.2	....
3	75	258.4	15.29	12.86	28.15	16.44	14.02	30.46	1,056.7	37.5	34.7
3a	45	258.4	15.22	12.77	27.99	.....	.....	.....	1,053.7	37.6	....
4	60	253.4	11.90	10.73	22.63	12.00	10.76	22.76	861.3	38.1	37.8
5	100	252.8	11.85	10.83	22.68	11.94	10.86	22.80	869.4	38.3	38.1
5a	40	252.5	11.90	10.88	22.78	.....	.....	.....	870.5	38.2	....
6	120	253.2	11.83	10.88	22.71	11.93	10.91	22.84	865.5	38.1	37.9
6a	60	252.9	11.79	10.83	22.62	.....	.....	.....	863.0	38.2	....
7	60	252.3	10.64	9.49	20.13	11.60	10.40	22.00	810.2	40.2	36.8
8	60	250.2	10.45	9.32	19.77	11.40	10.22	21.62	790.7	40.0	36.6

TABLE VI.

DATA OBSERVED IN TEST NO. 2.

Time. P.M.	Steam pressure at engine in lbs. per sq. inch above atmos.	Revolution Counter.		Weight of Water Barrels in Lbs.			Prony Brake Scales. Gross W'ght in Lbs.†	Tachometer.	Thermometer in Barrus calorimeter. Degrees Fahr.
		Total reading.	Difference.	Initial.	Final.	Net weight.			
8:35	72	48,428	....	....	....	....	190	158	221
8:40	68	49,723	1,295	....	....	....	190	158	222
8:45	75	51,020	1,297	....	....	....	190	158	226
8:50	73	52,320	1,300	....	....	....	190	158	233
8:55	73	53,610	1,290	79½	369½	289½	190	157	221
9:00	70	54,899	1,289	....	....	....	190	157	220
9:05	72	56,187	1,288	....	....	....	190	157	224
9:10	71	57,471	1,284	....	....	....	190	157	223
9:15	75	58,770	1,299	94½	381½	286½	190	157	222
9:20	70	60,056	1,286	....	....	....	190	157	222
9:25	73	61,336	1,280	....	....	....	190	157	225
9:30	70	62,609	1,273	....	....	....	190	157	238
9:35	73	63,902	1,293	80	365	285	190	157	224
9:40	75	65,190	1,288	....	....	....	190	156	232
9:45	73	66,476	1,286	....	....	....	190	157	222
9:50	72	67,752	1,276	....	....	....	190	157	239
9:55	71	69,034	1,282	95	379½	284½	190	157	237
Averages and Totals.	72.1	257.6*	1,288			1,146	190	157	227

* Revolutions per minute. † Initial weight on Prony brake scales = 55 lbs.

TABLE VII.

DETAILS OF MEASUREMENTS OF MEAN HEIGHTS OF INDICATOR CARDS FOR TEST NO. 2.

Time Card was taken. P.M.	Mean Heights in Inches. Head end. Actual.	Head end. Corrected for release.	Crank end. Actual.	Crank end. Corrected for release.	Time Card was taken. P.M.	Mean Heights in Inches. Head end. Actual.	Head end. Corrected for release.	Crank end. Actual.	Crank end. Corrected for release.
8:35	0.344	0.372	0.293	0.322	9:20	0.335	0.366	0.295	0.325
8:40	0.349	0.374	0.303	0.335	9:25	0.335	0.366	0.294	0.323
8:45	0.348	0.376	0.313	0.341	9:30	0.341	0.369	0.303	0.331
8:50	0.335	0.336	0.300	0.329	9:35	0.341	0.369	0.303	0.334
8:55	0.333	0.364	0.300	0.334	9:40	0.333	0.367	0.301	0.323
9:00	0.343	0.375	0.304	0.332	9:45	0.347	0.375	0.304	0.333
9:05	0.344	0.375	0.307	0.335	9:50	0.341	0.369	0.305	0.356
9:10	0.333	0.365	0.307	0.334	9:55	0.337	0.368	0.306	0.338
9:15	0.331	0.362	0.299	0.326					
Average						0.339	0.369	0.302	0.331

TABLE VIII.

CALCULATION OF EQUIVALENT SCALE OF INDICATOR SPRING CORRECTED FOR ALL VARIATIONS IN THE HOT SCALE OF THE SPRING. ORDINARY VALVE. SAME WORK AS FOR SPECIAL VALVE.

Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.	Division of card.	Area.	Mean Height.	Corre-spond'g scale.	Area × scale.
A1	0.35	1.10	58.6	20.51	B1	0.20	0.62	58.8	11.66
A2	0.31	0.97	58.7	18.20	B2	0.10	0.31	56.8	5.68
A3	0.21	0.65	58.8	12.35	B3	0.05	0.16	56.8	2.84
A4	0.16	0.50	57.3	9.17	B4	0.005	0.02	56.8	0.28
A5	0.12	0.37	56.8	6.82	B5	0.00	0.00	56.8	0.00
A6	0.09	0.28	56.8	5.11	B6	0.00	0.00	56.8	0.00
A7	0.07	0.21	56.8	3.98	B7	0.00	0.00	56.8	0.00
A8	0.04	0.12	56.8	2.27	B8	0.00	0.00	56.8	0.00
A9	0.004	0.01	56.8	0.23	B9	0.00	0.00	56.8	0.00
A10	0.00	0.00	00.0	0.00	B10	0.00	0.00	56.8	0.00
Totals...	1.354	.....	.....	78.64	Totals...	0.355	.....	.....	20.46

Equivalent scale = (78.64 − 20.46) ÷ (1.354 − 0.355) = 58.2

TESTS TO DETERMINE LEAKAGE OF VALVES.

Tests were made to determine the leakage of the two valves by placing them in the middle position, over the valve face, and weighing the steam which leaked by them when they were subjected to the full pressure. The amount of steam and water which passed through the ordinary valve was about 10 lbs. per hour, and through the special valve 2 lbs. per hour. No correction has been made for this leakage. An examination of the valves when subjected to the above tests indicated that a considerable portion of this leakage was in the form of water, which condensed from the steam and afterward leaked through the valve. It is, therefore, probable that the leakage when the valves were in motion was less than when tested as above.

DISCUSSION.

Mr. John H. Barr.—During the past year an investigation was conducted in the Sibley College Laboratory, under my direction, which has a bearing upon the subject of Professor Jacobus' paper, although the object of our work was somewhat different. Several months ago I became much impressed with the idea that, by a suitable system of compression governing, the economy of a steam engine under light loads might be considerably improved, and I devised a valve mechanism for varying the mean effective of the engine in this way. It has long

been held that the early exhaust closure in the single valve automatic engines is incidentally a source of economy. Dr. Thurston suggested, ten years ago (Trans. A. S. M. E., Vol. II., page 346), that variation of compression might afford an economical means of changing the M.E.P. Some time after beginning this work, I came across the paper of Mr. Tabor (Trans. A. S. M. E., Vol. V., page 48), which sustains this opinion; and many prominent members of the Society have agreed with this view. Our experiments were made upon the high pressure element of the Sibley College triple expansion Reynolds-Corliss engine. By changing the positions of the eccentrics, and the lengths of some of the rods, the engine was made to close the exhaust very early in the return stroke; and the excessive compression loop, that would otherwise have been formed at the end of the return stroke, was avoided by providing communication between the cylinder and the steam chest when the pressure in the former reached initial pressure. Fig. 257 of Professor Carpenter's paper on "The Saturation Curve as a Reference Line" (presented at this meeting), shows a reproduction of the diagram obtained in our trials with this steam distribution; and Fig. 256 shows the diagram obtained with substantially the same M. E. P. when regulating in the usual manner of the Corliss engine. It will be noticed that in the latter case the compression is moderately high for a Corliss engine, but rather less than is usual in the high speed engines. In the case of Fig. 257, the compression is abnormally high; and, furthermore, in this case, the free expansion loss at the end of the forward stroke is considerable, while the case shown in Fig. 256 is much nearer to the conditions demanded for economy by Mr. Ball's theory. We had expected that the distribution of Fig. 257 might reduce the cylinder condensation, and the comparison given in Professor Carpenter's paper indicates that this result was attained. However, the actual results of our trials show *almost the same steam consumption per horse-power per hour with either method.* There was a small gain with the smaller compression, but not at all commensurate with the difference which Professor Jacobus gives when the enormously greater variation of compression in our trials is considered. It appears from our work that the high compression method of regulation gave a better quality of steam and a higher average temperature in the cylinder, while it was accompanied by a compensating loss which we have not yet followed out. It may be that this loss is due to the departure

from the distribution which Mr. Ball gives as the proper one. It seems to me that one thing is rather clearly indicated by these trials, however, and that is as follows: Mr. Ball's conclusion, as I understand it, rests upon the assumption that the cylinder condensation is nearly a constant; our work indicates that this amount may be much reduced by compression, but that in the experiments quoted the compression may have been carried too far to secure a *net gain* in economy. It is quite evident that, with the compression system, we keep the steam in the cylinder at the higher temperature for a greater portion of the stroke, and at the lower temperature for a shorter time than when running with the corresponding M.E.P. under the more usual method of operation. This should, and probably does, reduce the actual amount of steam condensed per stroke. It would appear that this, if true, would affect the application of Mr. Ball's law; for an increase of the M.E.P., by adding to the forward pressure, tends to maintain a higher temperature in the cylinder; while a corresponding addition to the M.E.P., by reduction of the back pressure (through less compression), tends to lower the mean temperature of the cylinder. The waste per horse-power, through condensation, is proportional to $\frac{\text{condensation}}{\text{M.E.P.}}$; either a higher expansion line or a lower compression line increases the denominator of this expression; the latter change tends to increase the numerator as well, while the former method of changing the M.E.P. tends to reduce the numerator (or at least to increase it much less than a corresponding change in the back-pressure line), and hence it is more effective in reducing the *condensation rate.* Does it not appear then, even if we accept Mr. Ball's conclusion in a general way, that the free expansion at release should be somewhat greater than the free expansion of the boiler steam into the clearance space at admission?

We hope to make further experiments along the line of those quoted, with an engine better adapted to this kind of work; and, while there is not much prospect of realizing any direct commercial gain, it is hoped that the results may not be without value. It is possible that a fuller account of these trials, and of others which we have made and hope to make, may furnish the basis for a paper before the Society at some time in the future.

These trials were conducted with much care; the steam used was condensed in a surface condenser and weighed. Eight runs

were made (four under each system), and they were of from two to three hours' duration each. Indicator diagrams, gauge readings, etc., were taken every ten minutes, and the water measurement was taken at the signal for the other readings, and also when each tank was filled. The latter readings were, of course, the more reliable, although the sum of the partial readings checked the totals within a fraction of a pound. These partial readings were practically constant, showing a very uniform water rate (as should be the case under the steady brake load), and indicating that the water readings are entirely reliable.

The account of this investigation has been introduced in connection with the present paper, because the variation of compression was so much greater than in Professor Jacobus' experiments, and because our work indicates that there may still be, as has long been held by many, an important compensation in high compression.

Mr. F. H. Ball.—I quite agree with Professor Barr on some of the points he has made. I think, however, that I might suggest an explanation for the apparent difference between the results which he obtained, and those which Professor Jacobus found. I presume that, in the engine tested by Professor Barr, the clearance was very small, and we know that in the engine that Professor Jacobus tested the clearance was unusually large, and this difference in the engines would make quite a difference in the results.

One other point. Professor Barr concludes that the best economy would be obtained when compressing to a point somewhat higher than that which would be indicated by the theory that I suggested. I presume this would be true, for the reason he suggests, but I imagine that a careful test would show a very slight difference in that direction, because, for every change in the compression in the direction indicated, the point of cut-off would have to be made later, if the same mean effective pressure is to be realized, and, therefore, there would be more free expansion loss at exhaust opening, and that, of course, would offset any gain there might be in the heating of the cylinder. I will say further that Professor Barr has written me that he will conduct further tests on this subject, and I presume he will have interesting data for presentation at some later meeting.

*Professor Jacobus.**—The indicator cards referred to by Professor Barr are reproduced in Figs. 263 and 264. It will be seen

* Author's Closure, under the Rules.

that the conditions under which his tests were made are those not found in usual practice, as the engine was worked under very light loads. Mr. Ball's method of obtaining the proper compression curve is an approximate one, and is not true for extreme ranges in the load, and may not be true for any case where the load is an extremely light one. In the case of a light load his method would indicate that there should be full compression for an expansion line that meets the atmospheric line, or nearly a complete compression line in case the expansion line nearly reaches the line of atmospheric pressure. Starting with the conditions of Professor Barr's tests represented in Fig. 263, the proper diagram, according to Mr. Ball's analysis, would be about as represented in Fig. 265. Now, it may be true that the distribution obtained by Professor Barr, as represented in Fig. 264, may give an equal or better economy than the distribution indicated by Mr. Ball's theory, as represented in Fig. 265; but Professor Barr has not covered this point in his tests. Again, if full compression is found to produce a gain in economy at extremely light loads, it does not alter the conclusion drawn in my paper, which is for cases of everyday practice where the steam is cut off at about one-quarter stroke. The tests, then, show that there should not be full compression.

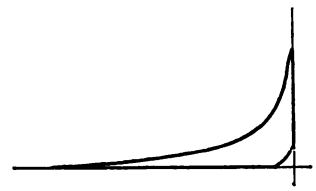

FIG. 263.

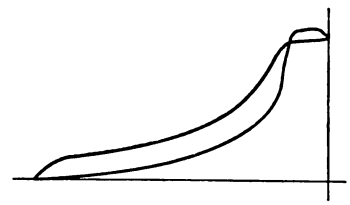

FIG. 264.

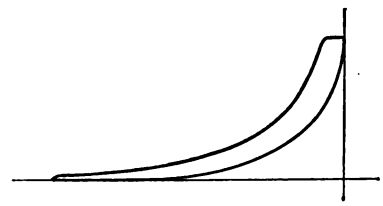

FIG. 265.

Mr. Ball suggests that the amount of clearance may cause a difference in the results obtained by Professor Barr and myself. This is probably so. In one set of Professor Barr's tests the steam was cut off so near the beginning of the stroke that, practically, only the clearance space was filled at cut-off, and for this case a difference in the clearance will probably produce considerable effect on the relative economy. My experiments show that in an engine of the high-revolution type, running under the practical

conditions adopted in the tests, there is a *very small* gain in economy for a given power in compressing the cushion steam to some pressure lower than the initial pressure, when the steam pressure is constant and the cut-off is varied to produce the required power. This gain is so small that it may be annulled by slight differences in the area due to variations in the points of release. This was the case in my experiments, 5% more steam being used per horse-power with the partial than with the full compression curve for a given power, when the areas of the indicator cards were not corrected for differences in the points of release. There is a larger gain in economy in favor of the lower compression when the point of cut-off is maintained constant at about one-quarter of the stroke, so that the power is different for the conditions of full and partial compression. This gain, in my experiments, amounted to 8.2% when the difference in the points of release was corrected for, or to 1.3% when no correction was made for the difference in the points of release.

DXCIV.*

NOTES ON THE THEORY OF SHAFT GOVERNORS.

BY ALBERT K. MANSFIELD, SALEM, OHIO.
(Member of the Society.)

AT our spring meeting of 1890, at Cincinnati, three papers on shaft governors were presented, which, together with the extended discussions thereon, make a valuable contribution to the literature of this subject.

It is the purpose of this paper to add something to that discussion, with the hope of getting a little nearer to a correct solution of the perplexing problems surrounding the subject.

The matter to be discussed will be divided into several topics, for the sake of clearness.

1. *The path of an unbalanced governor ball of a shaft governor, isochronously adjusted.*

This problem was proposed by Professor Sweet in one of the papers referred to, and in an elaborate mathematical analysis Prof. S. W. Robinson seems to prove that the centre of the approximately circular path is vertically above the centre of the shaft, and that the distance apart of these centres may be determined from the formula $h = \frac{9.78}{n^2}$; h being the distance sought, in inches, and n being the number of revolutions per *second.*

Confirmatory of this result, an experiment was made with an unbalanced weight, arranged according to the requirements of the problem. The figure illustrating the experimental apparatus is here reproduced (see Fig. 266).

The spring adjustment was such that this apparatus was thought to be isochronous at 555 revolutions per minute, and the experimental determination gave $h = 0.126$ inch (mean of

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions.*

several trials), while by the formula h becomes 0.114 inch—a fairly close agreement.

Following this a table of values of h was given, derived from the formula, revolutions per minute being taken at from 1,200 as a maximum down to 60; h in the former case being by calculation 0.02 inch, and in the latter 9.78 inches.

Suppose this table to be extended to cover slower speeds than sixty revolutions per minute, even down to one revolution per minute, h would be found at this speed to be, by the formula, 35,200 inches, or 2,933 feet—more than half a mile above the centre of the shaft.

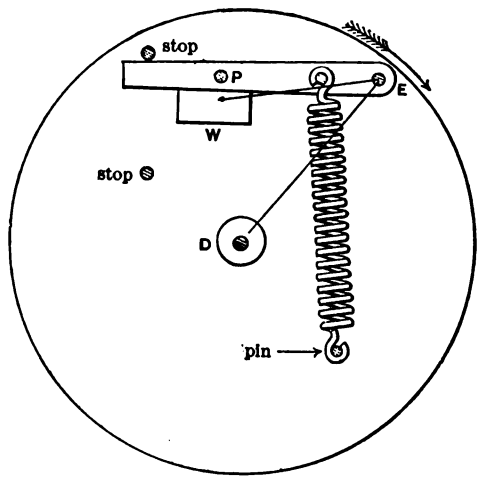

FIG. 266.

This extreme result is noted merely as a curious matter of interest.

The formula is doubtless correct, as deduced from the assumption on which it is based; but let us examine the assumption.

Referring to the figure of Professor Sweet's paper, here reproduced (Fig. 267), Professor Robinson says: "Suppose, to start with, that the weight B is at j, moving along a horizontal portion of arc. The action of gravity tends to deflect it downward instead of allowing it to move along a circular path concentric with the shaft, thus giving the mass B an accelerated motion relative to the wheel A, and along the radius AB, so that by the time the weight reaches a it will have a considerable

velocity toward the centre C. From this point on gravity counteracts, and on reaching d will have destroyed the radial velocity toward C, when B will again be moving horizontally, or perpendicular to the radius, but will be at a point nearer the centre C than when at j. Now, from this point on a radial acceleration will occur, so that at g the weight will be moving outward with a radial velocity which, from g on to j, will again be destroyed by gravity, thus bringing the weight to rest on the radius at j, though at a greater radial distance from the centre C than at any point before in the revolution, and putting the weight in the position and condition supposed at the start, when it will go on in repetition of the curve as the next turn of the shaft is made, and so on, continuously, the curved path described being found to be nearly a circle, with its centre elevated above that of the shaft."

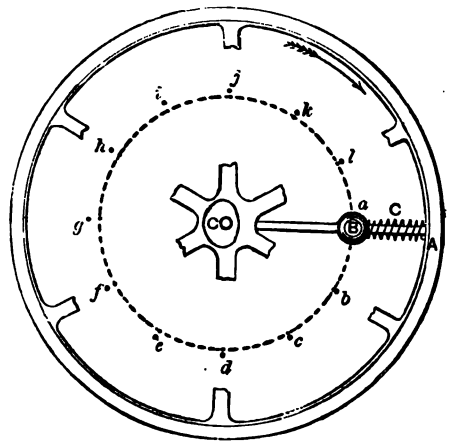

FIG. 267.

The questionable part of this reasoning lies in the first sentence: "Suppose, to start with, that the weight B is at j, moving along a horizontal portion of arc." Under this supposition what follows is justified; but the supposition is only one of an indefinite number that may be made with equal correctness, each leading to a different conclusion.

For example, suppose we start the analysis from the point a, the weight being assumed to have no radial movement at that point, and the motion to be right-handed. Then, during the entire first half-revolution, an accelerating force is drawing the weight away from the centre, always faster, until at g this force becomes zero, and the velocity outward is uniform. During the next half-revolution, a force of equal effect acts to draw the weight toward the centre, but this force must be entirely expended in overcoming the velocity of the weight outward, which it had when at the point g, therefore the movement of the weight is outward during the whole revolution. The same action occurs

in succeeding revolutions, and the weight describes a spiral outward, finally reaching its outer stop.

If we start the analysis from g, we find by similar reasoning that the weight describes a spiral inward.

The reasoning in the case of starting from g or a, which does not permit the ball to return to its starting point, is found to be rational, when compared with the case of any weight moved in a straight line, with no resistance except that of inertia, by an accelerating force, and stopped by an equal retarding force. The weight comes to rest, and no work is gained or lost, yet the weight is found in a new position.

Again, if we consider the weight to start, with no radial movement, from any other points intermediate between its positions, a, g, and j, d, it will be found to describe a spiral outward when the first position is taken at the right of the axis, and inward when at the left. These spirals are, as will be seen by consideration of the forces of inertia, in no case regular spirals, but are merely of spiral nature, not re-entering.

The conclusion to be derived from this analysis seems to be that the problem has no true solution, or if any expression based on correct reasoning could be found for the curve, it would be irrational.

Moreover, under the conditions of perfect isochronism assumed, one would be led to expect the motion of the ball to be erratic; as, for instance, it might first move in the outer spiral, when, reaching the outer stop, it may be compelled to move horizontally at the point j, which might start it in the eccentric circle. Slight disturbing influences, as of the atmosphere, would probably change it from this to other of its paths. This expectation seems to be to some extent verified by the experiment of Professor Robinson, for we find in his table of results the remarks "spiral inward," "spiral outward," "steady," etc.

The practical conclusion to be derived from this analysis is that an unbalanced weight in an isochronous shaft governor is not feasible.

2. *Centrifugal moment of a governor weight.*

Theorem: The centrifugal moment about the weight pivot, like the simple weight moment, is the same wherever on the

line through the centre of gravity of the weight to the centre of the pivot the weight be considered as concentrated.

Let A, Fig. 268, be the centre of the shaft, B the centre of gravity of the weight, and C the pivot centre or fulcrum.

In considering simple weight leverage about C, the effective weight may be assumed to act at any point on the line $B\,C$, as at

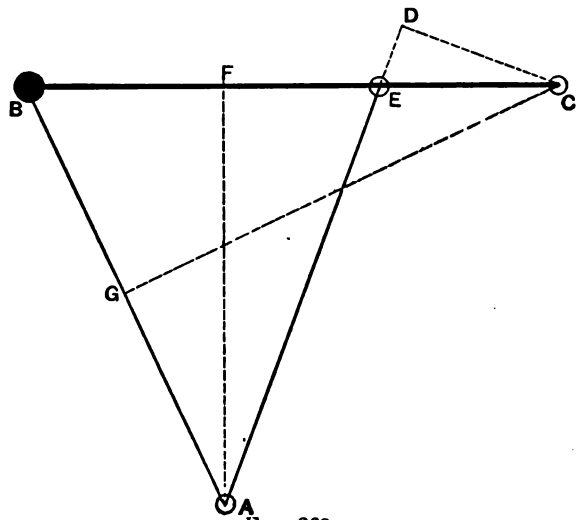

FIG. 268.

E, in which case the weight at E is to the actual weight at B as $B\,C$ to $E\,C$. The centrifugal moment of weight B for any given number of revolutions is proportional to the radius $B\,A$ times the line $G\,C$, drawn at right angles to $B\,A$ through C. The centrifugal force of the resultant weight E is similarly $A\,E$ times $D\,C$; $D\,C$ being drawn at right angles to $A\,E$ prolonged.

According to the theorem,

$$A\,B \times G\,C \times C\,E = A\,E \times D\,C \times B\,C,$$

the lines $C\,E$ and $B\,C$ being the relative weights in the two cases.

Proof: Draw the line $F\,A$ at right angles to $B\,C$. Then we have the similar triangles $A\,F\,B$ and $C\,G\,B$; also $A\,F\,E$ and $C\,D\,E$,

from which $\quad A\,B : A\,F = C\,B : C\,G,$

and $\quad C\,E : C\,D = A\,E : A\,F.$

Multiplying the equations together, and equating product of means to product of extremes, we have

$$A\,B \times G\,C \times C\,E = A\,E \times D\,C \times B\,C$$

as asserted.

This goes to show that the attempt, which is evident in some shaft governors, to so design them that the centre of gravity of the weight shall move nearly in a radial, line is unnecessary. Wherever the weight be concentrated on the line $B\,C$, provided its amount be sufficient, the result is the same whether the arc described approaches a radial line or not.

This demonstration leads up to another interesting detail, which seems to show that shaft governors are not always arranged for true isochronism.

3. *Position and tension of spring.*

In Fig. 269 let A be the centre of shaft, B the centre of gravity

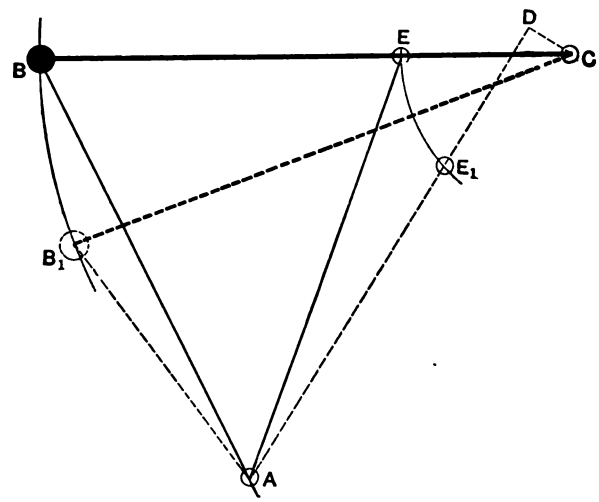

FIG. 269.

of weight, and C the weight pivot, as before. Since the line $A\,B$ may be taken as a measure of centrifugal force of B, then a line from A to B at any other position of B in its arc will be, in an isochronous governor, assuming uniform velocity, a corresponding measure of the centrifugal force of B in this new position,

as the line $A B_1$ for the position B_1. Suppose a pull spring pivoted so as to swing about the point A, and to be pivotally attached at its outer end to the centre of gravity of B; also, suppose the spring to be of such tension and strength as to exactly balance the centrifugal force of B. Suppose, also, that the line $A B$ represents the total extension of the spring—*i. e.*, that when the end B of the spring is at A the force of the spring is zero—then the spring, from the laws of spring tension, will just balance the centrifugal force of the weight B in other positions, as at B_1. The arrangement is therefore isochronous, for centrifugal and centripetal forces are exactly opposed to each other in direction and amount.

Take any point E, as before, on the line $B C$, and draw the line $A E$; consider $A E$ to represent a spring pivotally supported at A and pivotally connected at E, and of such force as to counterbalance the centrifugal force of B, or of its resultant weight at E. Then, if the zero of tension of this spring is at A, it will counterbalance correctly the centrifugal force of the weight at all other positions, as at E_1, for the spring lies in the line of action of centrifugal force, and its leverage $D C$ about C is the same as that of the centrifugal force. From which it is clear that a spring pivotally adjusted at A, and pivotally connected to the weight-arm *at any point* on the line $B C$, is correctly located to produce exact isochronism.

In Fig. 270 let A, B, B_1, and C represent shaft, weight, and pivot, as before. Draw lines $A C$ and $B_1 A$. Draw an arc from C through any point E on the line $B C$, cutting $B_1 C$ at E_1. From the point E draw $E A_1$, parallel to $B A$, and connect A_1 to E_1. The figure $A_1 E E_1$ is exactly similar to the figure $A B B_1$, and corresponding sides of the figures are parallel to each other. It will be clear from the foregoing and from inspection, that if a spring be pivotally connected from A_1 to E, and has its zero of tension at A_1, and is adjusted to balance the centrifugal moment of the weight, it will balance it in all other positions, as at E_1. But the arc $B B_1$ need not have been drawn through the centre of gravity B, for, from the previous demonstration (see Fig. 268), it could have been drawn through any other point of line $B C$; therefore, the line $B A$ might have had any direction between that of the direction of the line $A C$ to that of the direction of a line $A B_2$, parallel to $B C$. Such an indefinite number of constructions would bring the point

A_1 at any position on the line $A\ C$, or its continuation through A.

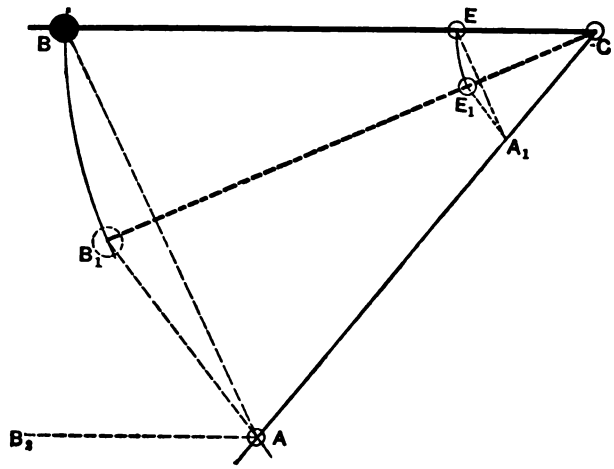

FIG. 270.

It has therefore been shown that *a spring pivotally swung at any point on the line $C\ A$, or its continuation through A, and pivotally connected to any point on $C\ B$ or its continuation through B, and*

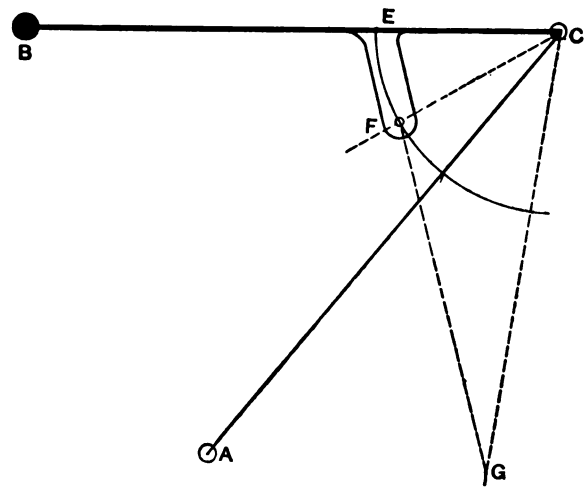

FIG. 271.

having its zero of extension at the first-named point, is correctly placed to produce exact isochronism.

Referring to Fig. 271, it will be clear without demonstration

that the point of connection to weight-arm need not be on the line $B\,C$. It may be anywhere on the weight-arm, as at F, provided a new zero line $G\,C$ be drawn angularly the same distance and direction from $A\,C$ as the line $F\,C$ from $E\,C$.

It follows that the point of connection of spring to weight-arm, and the direction of action of spring, *may be selected entirely at random*, or for convenience, provided only that the length between pivots and the tension of spring be fixed according to the principles laid down.

The conclusions arrived at by the preceding reasoning may be expressed in the form of a second

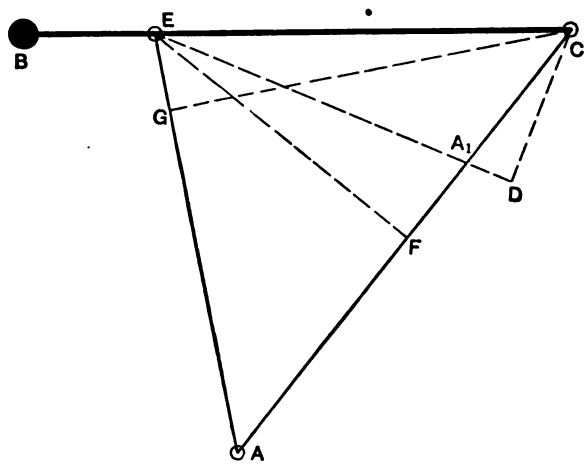

FIG. 272.

Theorem: The combined zero and fixed pivotal point of a spring, arranged to act isochronously on any point of the line of weight-arm from weight pivot through centre of gravity of weight, may be taken at any point on the line from weight pivot through centre of shaft. Moreover, the spring force required will be inversely as the distance of the fixed pivot from the weight-arm pivot.

In Fig. 272, letters $A\ A_1\ B\ C\ E$ represent the same parts as in Fig. 270. Drop perpendiculars $C\,G$ and $C\,D$ from C on lines $A\,E$ and $E\,A_1$ respectively; also from E drop the perpendicular $E\,F$ on line $A\,C$. Then similar triangles $A_1\,E\,F$ and $A_1\,C\,D$, as well

as AFE and ACG, are formed, from which proportions may be made as follows:

$$AE : EF = AC : CG,$$
$$A_1C : CD = A_1E : EF.$$

Multiplying together, equating products of extremes and means, and canceling EF, we have

$$AE \times CG \times A_1C = A_1E \times CD \times AC,$$

or

$$AE \times CG = A_1E \times CD \times \frac{AC}{A_1C}.$$

If we consider AC to be the unit of force of the spring when the fixed pivot is at A_1, and A_1C the unit of force when the pivot is at A, then this becomes intelligible.

It shows that the linear extensions of the springs AE and A_1E, multiplied by their leverages GC and CD and by the units of force of the springs, are equal. The points E and A_1 were taken at random, which makes the demonstration general.

A further consideration of Fig. 268 will show that when A is the fixed pivot the unit of spring force is inversely as the distance of the point of connection on line BC from C.

If, therefore, we have computed the centrifugal force of the weight B, we have merely to multiply this centrifugal force by the ratio $\frac{BC}{EC} \times \frac{AC}{A_1C}$ (see Fig. 270 for illustration) to find the corresponding balancing spring force; or to multiply the centrifugal force per inch of radius by this ratio to find the corresponding spring force per inch of extension. The linear extension of the spring was before shown to be EA_1.

4. *Approximate isochronism.*

In Fig. 273 let ABC be the centres of shaft, of gravity, of weight, and of pivot, as before. Draw a line from B through A to any point O. Let O be the fixed point of a spring pivotally attached to B, and having its zero of extension at A.

It is clear that the arrangement may be made isochronous for the two positions A and B. For the moment consider O to be infinitely removed from A, and investigate the mid-position of B at B_1. Draw B_1O_1 parallel to BO, and AA_1 parallel to B_1C; also AB_1, and DC at right angles to AB_1 through C.

Let $A\,C$ be represented by R and the angle $A\,C\,B_1$ by a. Then the spring moment at B_1 is $R^2 \sin a$, and the centrifugal moment is $2\,R^2 \sin \frac{a}{2} \times \text{cosin} \frac{a}{2}$, which expressions are equal to each other, by trigonometry. Therefore, with a spring so located and adjusted, a third point, B_1, is isochronous. If the point O is a finite distance from A, as shown, it will be found that there will still be a point B_1, near the middle of the arc $A\,B$, which

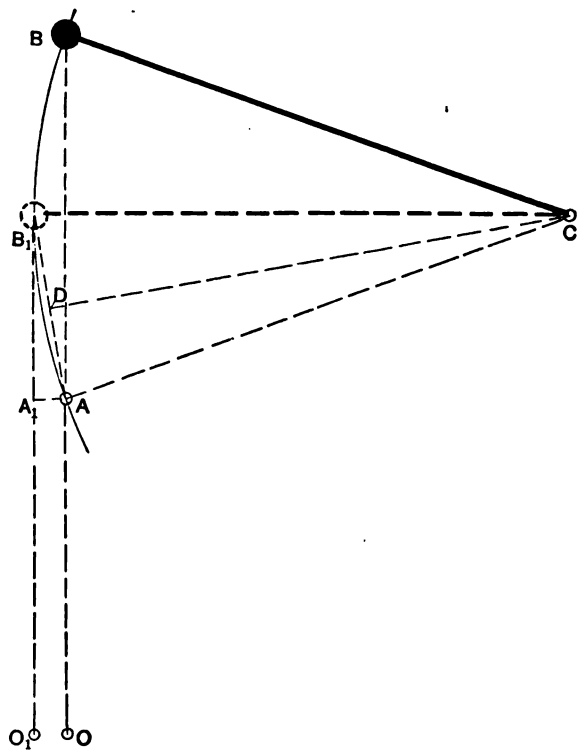

FIG. 273.

will be isochronously balanced; other points, however, between A and B_1 will have their spring moment too small, and points between B_1 and B will have their spring moment too large.

Suppose the spring to be arranged as in Fig. 274, the point B_1 being the inner or initial position of the weight B; then clearly, from previous demonstrations—O being the pivotal point of the spring—to produce isochronism at points B_1 and B, A_1 must be the zero point of the spring. It will be found as before that a

point nearly midway between B and B_1 is also isochronous; also, if the angle $B\,C\,B_1$ is not large, the approach to complete isochronism is very close.

This corresponds to the arrangement commonly used in practice. Clearly, the arc $B\,B_1$ may be drawn in any other place from C as a centre, as at $D\,D_1$, the line of the spring being made to pass through these two points, and their angular distance apart being the same as that of B and B_1.

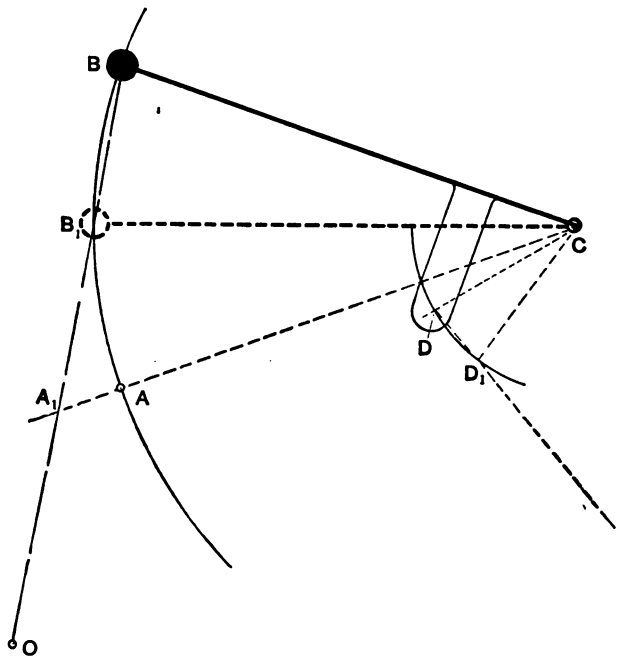

FIG. 274.

5. *Influence of the weight of the spring.*

Let the spring be applied, as in Fig. 275, and let G be its centre of gravity. Determine its moment m of centrifugal force about its pivot O, and divide by the length of the spring $D_1\,O$, which we will call l. Owing to the weight G being constant, and the direction of l practically always the same, $\frac{m}{l}$ is very nearly constant—m being the moment of centrifugal force and l the length of the spring—for any degree of extension between D_1 and D, and the quantity $\frac{m}{l}$ is the tangential force at D, due to the cen-

trifugal force of the spring. Extend the arc $D\,D_1$ across the line $A\,C$, and lay off each side of the line to F and F_1 one-half the arc $D\,D_1$. Draw radial lines from C to F and F_1. Draw a circle through C whose diameter is the chord $D\,D_1$. From A draw lines tangent to this circle, crossing the radial lines $C\,F$ and $C\,F_1$ at K and K_1. Let d represent the distance $A\,K$.

Then, since $\frac{m}{l}$ is the force, $\frac{m}{ld}$ is the weight, which, concentrated at K or K_1—according to the position of the weight-arm

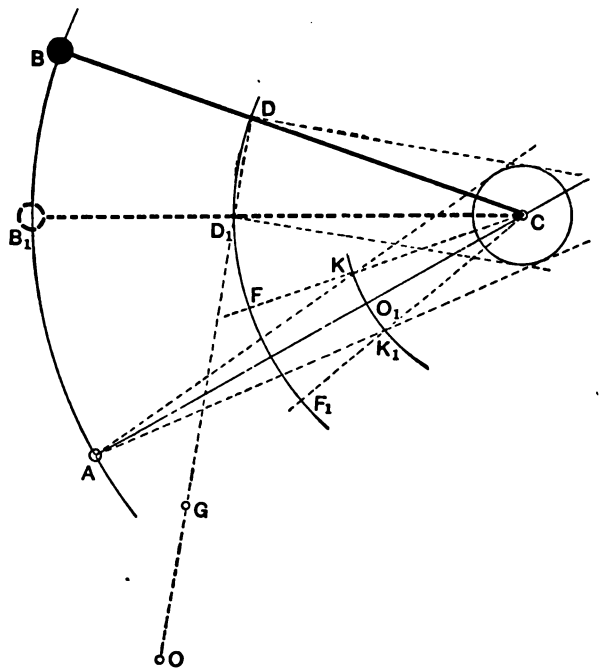

FIG. 275.

$B\,C$—will produce a centrifugal moment about C almost exactly equivalent, at the three points K, K_1, and O_1, to the centrifugal moment due to the spring weight. Between the extreme positions K and K_1 and the central position O the action of such a weight is not exactly equivalent to that of the spring, but with a moderate arc the error is extremely small.

The object of determining the location and amount of the equivalent weight K is to find the influence of the spring on the location of a correct centre of gravity line $B\,C$.

6. *Influence of the weight of the link.*

Theorem: If a governor link be constrained to move at one end in a circular path about the centre of the driving shaft, and at the other end in a circular path about the centre of the weight pivot, then is the centrifugal effect of the link the same as if that portion of the weight of the link were concentrated at its weight-arm pivot, which would rest on its support if the link were placed in a horizontal position on two end supports.

In Fig. 276 let A be the shaft centre, C the weight-arm pivot, $L\,O$ the link, and G the centre of gravity of the link. $G\,A = G\,A_1$ may be taken to represent the centrifugal force of the

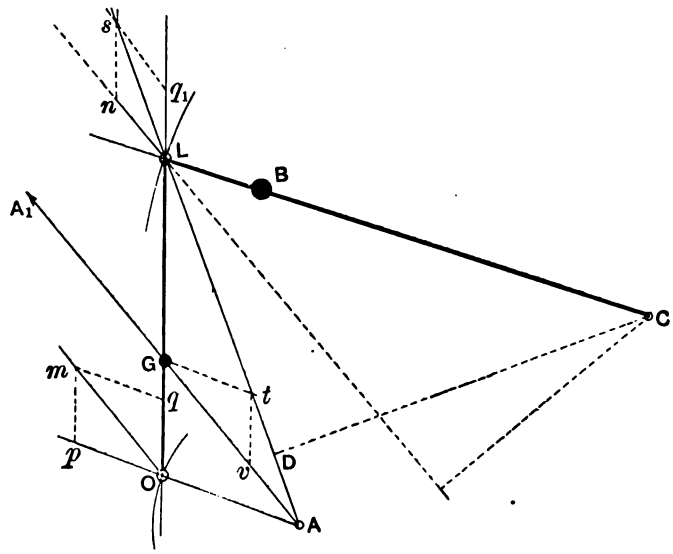

FIG. 276.

link. Clearly, this force may be resolved into two forces, $O\,m$ and $L\,n$, the sum of which is equal to $G\,A$, while their ratio is as $G\,L$ to $G\,O$, and their direction of action parallel to $G\,A$.

Resolve $O\,m$ into the components $O\,p$ and $O\,q$, $O\,p$ having a radial direction from A, and $O\,q$ lying in the line of the link. Lay off $O\,q$ from L to q_1, in the direction of the link, and combine $L\,q_1$ with $L\,n$ by the parallelogram of forces, which gives the force $L\,s$ as the total resultant force of the link tending to rotate the weight-arm about its pivot.

Draw $L\,A$, also $G\,t$ parallel to $O\,A$, and $t\,v$ parallel to $O\,L$.

By geometry $G\,A$ is divided at v, and $L\,A$ at t, in the same ratio as $L\,O$ at G; therefore, since $O\,m$ was made equal to $G\,v$ in amount and direction, the triangles $m\,O\,q$ and $G\,v\,t$ are equal. $L\,n$ was made parallel to and equal to $v\,A$, and $n\,s$ is parallel to and equal to $v\,t$ by construction; therefore the triangle $n\,s\,L$ is equal to the triangle $v\,t\,A$, and $L\,s$ is equal to $A\,t$ in amount and direction.

But $$\frac{A\,t}{A\,L} = \frac{O\,G}{O\,L}, \quad \text{or,} \quad A\,t = A\,L \times \frac{OG}{O\,L}.$$

In other words, when the centrifugal force acting at G is represented by the radius $A\,G$, that acting at L may be represented by the radius $A\,L$, multiplied by a fraction which is the ratio of the weight G, which would be supported at L, provided the link were to rest in a horizontal position on two supports at L and O; which was to be proved.

If the centre of gravity of the link were at its centre, as is common, then it would be exactly right to consider one-half the link concentrated at L.

7. *Frictional effect of valve.*

In Fig. 277 let $A\,C$ represent the maximum tension of spring and $B\,C$ the tension to inner position of weight-arm. Let $A\,D$

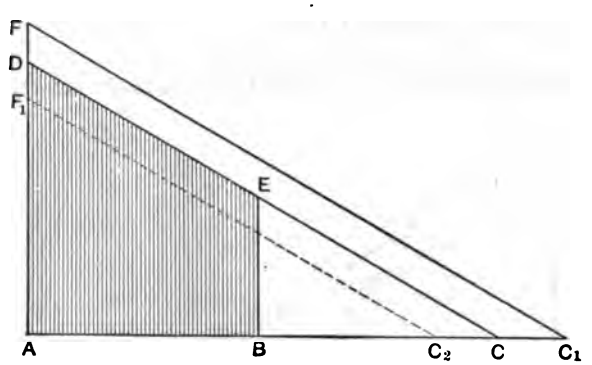

FIG. 277.

and $B\,E$ represent the spring force corresponding to positions A and B of weight-arm. Assuming perfect isochronism between weight and spring, then $A\,D$ and $B\,E$ also represent the balanced centrifugal force, and this force for any intermediate posi-

tion of weight-arm is the corresponding height from the line $A\,B$ to the line $D\,E$. Up to this time we have neglected the effect of valve-gear friction.

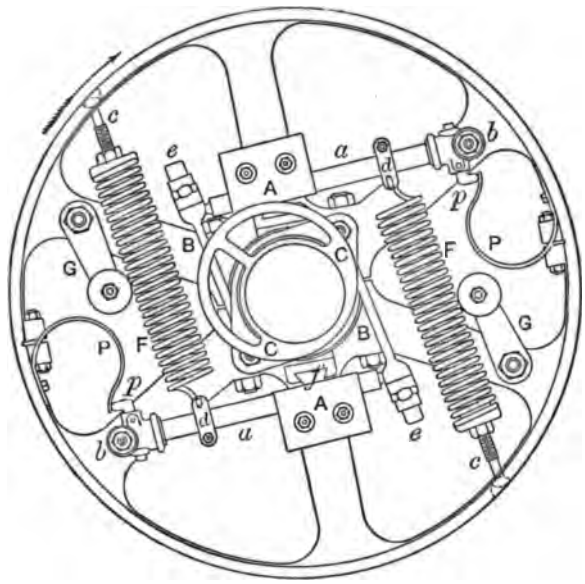

FIG. 278.

Supposing this effect to be a constant force acting in the same direction as the centrifugal force of the weight, then it may be represented by a line parallel to and above $D\,C$, as $F\,C_1$. If the constant friction of the valve-gear acts against centrifugal force,

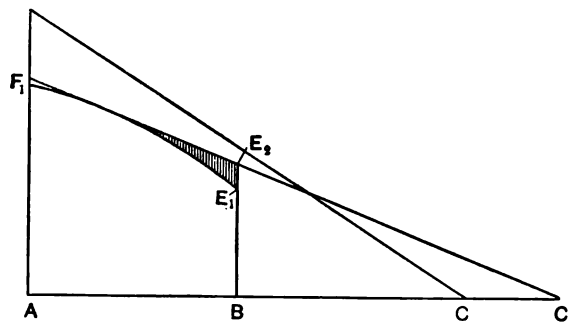

FIG. 279.

or with the spring, then $F_1\,C_2$, parallel to $D\,C$, may represent its effect. In the former of these cases the maximum spring force

becomes $A\ F$, and the maximum spring tension $A\ C_1$, while in the latter case these quantities become $A\ F_1$ and $A\ C_2$.

In some constructions the connection between the governor and the valve-gear is such as to produce a variable effect of friction. This is the case with the "Buckeye" governor, shown in Fig. 278, in which, moreover, this friction has a centripetal effect.

In Fig. 279 this variable resistance is illustrated by the curved line $F_1\ E_1$.

By reference to Fig. 278 an auxiliary spring P will be seen, which is designed to act through a little more than half the range of the weight-arm, and to produce an effect illustrated by the shaded portion of Fig. 279. The result is that a line $F_1\ C_1$ approximately straight, illustrates the centripetal action of the main spring, $B\ C_1$ being its initial tension, and $B\ E_2$ its total force at first stop, or initial position.

8. *Inertia in a shaft governor.*

In Fig. 280 let A and C be the centres of shaft and weight pivot, respectively, and consider the total effective weight of the

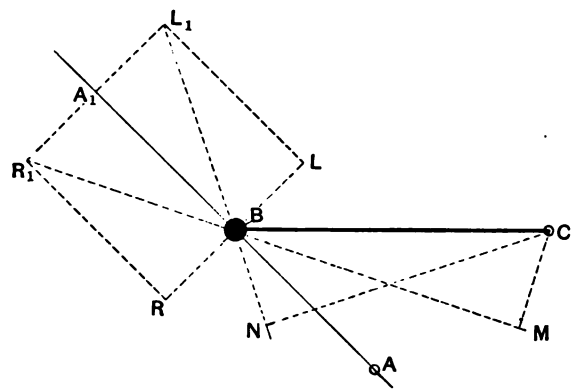

FIG. 280.

governor weight and arm centred at B. Inertia acts on the weight B at right angles to the line $A\ B$.

From analytical mechanics (see Weisbach) the *force* of inertia may be represented by the expression $P = M\ R \frac{d\ \omega}{d\ t}$, while centrifugal force is $F = \omega^2\ M\ R$; in both expressions ω is the angular velocity, M the mass, R the radius $A\ B$, and $d\ t$ the small interval of time in which a change of velocity occurs.

Substituting for ω its equivalent value $2\pi T$, in which T is the number of revolutions per second, and differentiating the equation for centrifugal force, since it is only the difference of force due to change of speed which is effective, we have

$$P = 2\pi M R \frac{dT}{dt},$$

and

$$dF = 8\pi^2 M R T dT.$$

If BL or BR, according to the direction of motion, represents the force P, and BA_1 the force dF, then BL_1 or BR_1 is the resultant of these forces, and the tangent of the angle $A_1 B L_1$, or $A_1 B R_1$, which angle we designate by α, is

$$\text{tangent } \alpha = \frac{P}{F},$$

or, substituting above values,

$$\text{tangent } \alpha = \frac{1}{4\pi T dt}.$$

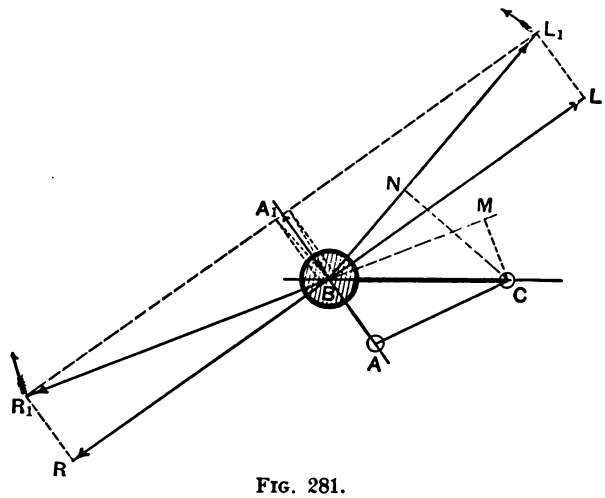

FIG. 281.

We see from this that the effect of inertia to increase or decrease (according to the direction of motion) the moment of force about the weight pivot is less the greater the number of revolutions per unit of time, and is greater the less the interval of time in which the change of speed takes place.

Let us assume that the weight B is no longer concentrated in a point, but is spread out into a disk of considerable size, as in

Fig. 16, whose radius we call r; then the force of inertia relative to the axis A is greater than before.

By a well-known law of inertia, the radius of gyration of the weight is $R + \frac{r}{\sqrt{2}}$, therefore the force of inertia acting at B is

$$P = \frac{\left(R + \frac{r}{\sqrt{2}}\right)^2}{R} M \frac{d\,\omega}{d\,t}.$$

Substituting $2\,\pi\,T$ for ω, and dividing by $d\,F$ as before, we have

$$\text{tangent } \alpha = \frac{\left(1 + \frac{r}{R\sqrt{2}}\right)^2}{4\,\pi\,T\,d\,T}$$

Suppose r to be $\frac{1}{3}$ of R, T to be three revolutions per second, and $d\,t$ to be one second; then tangent α becomes 0.0406, and α is less than $2\frac{1}{2}°$. If $d\,t$ is $\frac{1}{10}$ of one second, then α becomes about 22°, and if $d\,t$ is $\frac{1}{100}$ of one second, α is about 76°. The extremes of these three cases are shown graphically in Fig. 281 for both right and left hand motion. This illustrates to how great an extent, when changes of speed are sudden, inertia force may be useful to assist centrifugal force; also to what a slight extent inertia acts when changes are not sudden.

It also shows that if the direction of motion be badly chosen, the combined forces may produce an instantaneous moment about the weight pivot in the wrong direction, thus interfering with sensitive governing.

As to the actual value of $d\,t$ in practice, it may often be a very small quantity, for in an engine having dead points the velocity changes a number of times, to a greater or less extent, during each revolution. These changes are less the heavier the fly-wheel, therefore with a light fly-wheel an inertia governor should be specially efficient.

In one of the papers referred to at the beginning of this article, Mr. Armstrong advocates the use of inertia in the way which would reduce the effective moment about C, for the sake of "stability."

The fact seems to be, however, that stability and sensitiveness are best arrived at by using the force of inertia to aid centrifugal force, as in the left-hand motion of Fig. 281.

Fig. 282 illustrates a governor for a single-valve engine—de-

signed by Mr. J. W. Thompson—which is said to have performed so perfectly that no perceptible variation of speed in the range of the governor could be detected by careful test, and there was no trouble from racing.

It will be noted that the arc through which the weight-arm moves is so small that isochronism could be practically perfect, while inertia was utilized to a great degree.

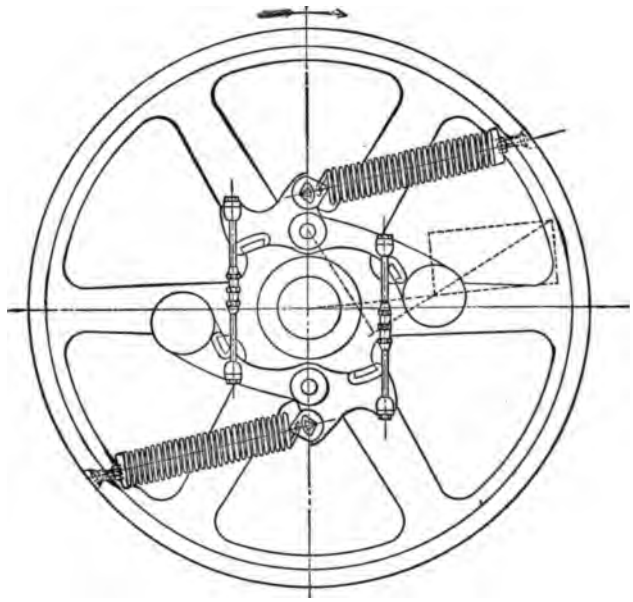

FIG. 282.

In Fig. 283 is represented a very ingenious method of combining a separate inertia weight with a shaft governor. This was applied by Messrs. Bancroft & Lewis, of Philadelphia, to a Buckeye engine in the works of Wm. Sellers & Co.

The inertia weight consists of a wheel, which, being centred on the shaft, has its centrifugal force completely balanced, while its inertia force acts to *aid* the centrifugal force of the governor. Instead of causing racing this is said to have overcome all tendency to race, thus enabling the governor to be adjusted for practically perfect isochronism.

It is well known that an increase in the amount of the balanced forces—centrifugal and centripetal—of a governor tends to increase the effectiveness of the governor to overcome dis-

turbing influences; yet an increase in these forces may produce an increase of friction in the pivots, which may defeat the desired object. The friction of pivots is not increased, however, by so designing the governor as to utilize inertia to aid centrifugal force.

In this respect the shaft governor may have a decided advantage over the old ball governor, which is purely centrifugal.

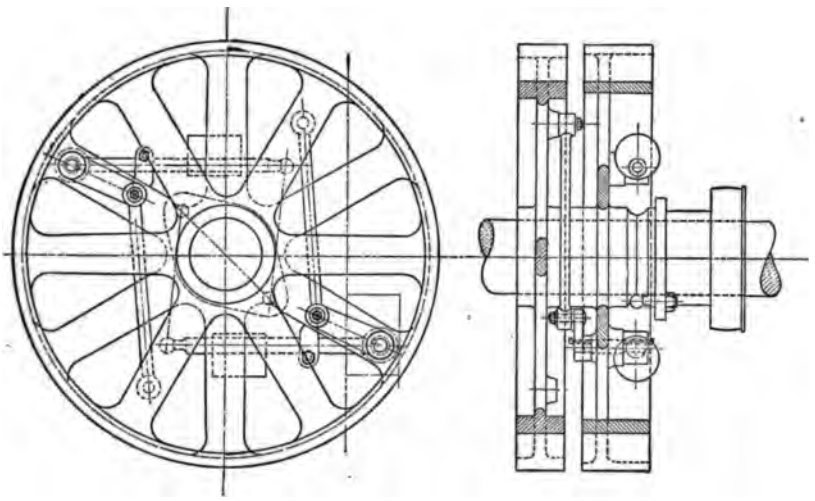

FIG. 283.

Referring back to Fig. 283, it is not essential that the inertia weight be centred on the shaft. It may be centred at the weight-arm pivot, thus forming a part of the weight-arm. If its centre of gravity is coincident with the centre of pivot it will not affect the centrifugal adjustment of the governor weight, but will aid the governing moment by its inertia.

DISCUSSION.

Prof. Gaetano Lanza.—I should like to ask Mr. Mansfield two questions. One is whether he knows of any experiments to show how the friction varies in a shaft governor, and how much it is, whether it is constant, or whether it varies by any law, or by no apparent law.

The other question is whether he has considered what is the effect produced by the bending of the springs in consequence of the centrifugal force of the springs themselves.

Mr. Mansfield.—I would say, in answer to the first question, that I do not know of any experiments with a shaft governor that would answer it reliably; but there are adjustments that may be made that seem, in my experience, to make it unnecessary to answer it, for we are able, by making the adjustments carefully, to produce almost perfect governing.

As to the bending of the spring through the centrifugal force, in engines as ordinarily constructed, which run at a uniform rate of speed—or as designed to—the bending of the spring must be nearly constant, and that merely changes the direction of action of the spring. If we were able to calculate with certainty as to just the amount of deflection of the spring, it should modify slightly the design of the governor; that is, what appears to be the direction of the action of the spring is not actually the direction when the governor is acting at the speed of the engine. Further than that I cannot answer the question from any experimental knowledge.

Professor Jacobus.—Mr. Mansfield has given us a theory of the governor, but in no place has he stated the actual time for a governor to move from one position of adjustment to another, and the relative time in moving, say from half load to no load, and from full load to no load. This was made a subject of experimental investigation at the Stevens Institute of Technology on a Ball & Wood engine, in which the action of the governor depends but little on the so-called inertia effect. I have obtained curves which represent the spiral path travelled over by the governor balls as they move from one position to another under various conditions. In the actual experiment the spirals are not taken directly from the governor balls, but they are taken from the centre of the valve eccentric, which, being moved by the governor itself, represents the move-

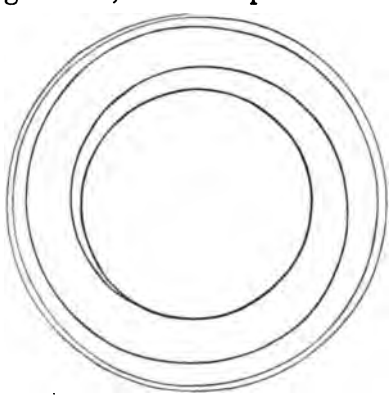

FIG. 284.—Springs at full theoretical tension. Revolutions 285 per minute. Dash-pot removed and governor in unstable equilibrium.

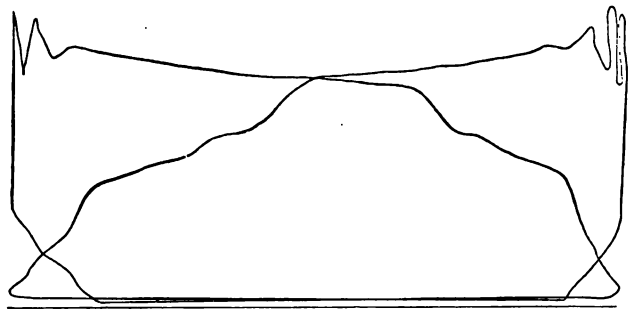

FIG. 285.—Indicator cards taken before removing the load and tracing the spiral shown in Fig. 284. Scale = 40.

ment of the governor after proper corrections are made. Figs. 284, 286, and 288 show these spirals.*

The indicator cards shown in Figs. 285 to 294 represent the steam distribution before and after the load is thrown off. By counting up the number of the coils of the spirals, we obtain the exact number of revolutions required for the governor to move from one position to the other, and in this way we find the amount of time required to make an adjustment.

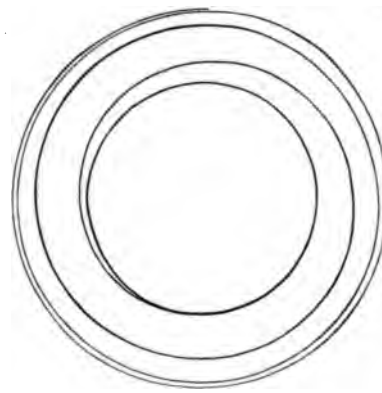

FIG. 286.—Tension of springs adjusted so that the governor is just at the point of unstable equilibrium. No dash-pot. Revolutions 265 per minute.

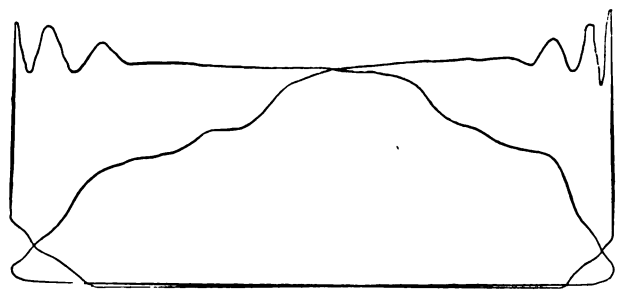

FIG. 287.—Indicator cards taken before removing the load and tracing the spiral shown in Fig. 286.

* The method of performing the experiments was described at the last meeting of the American Association for the Advancement of Science, and specimens of the curves obtained were exhibited.

There were two sets of curves obtained in the experiments, one with a dash-pot on and one with the dash-pot off. For the present discussion we will only consider those obtained with the dash-pot removed, which are those represented in the figures. The curves show that the time to effect the adjustment from one-half load to no load is the same as from full load to no load, and that the adjustment is accomplished in three and one-half revolutions. If the governor is set just at the point of "hunting," so that the springs are at the full theoretical tension, the time required to adjust from one load to another is about the same as the time required when the springs are set so that the revolutions are about forty per minute below the speed for correct adjustment. In other words, the governor will regulate just as quickly when the springs are set so that it is in a very stable condition as when set so that the governor is just at the point of unstable equilibrium.

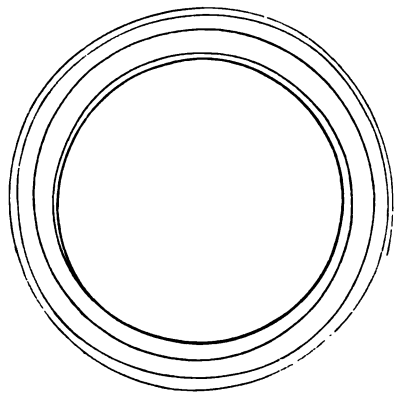

FIG. 288.—Adjustment of governor same as in Fig. 286, with less variation of load than that corresponding to Fig. 286. See indicator cards Nos. 287 and 289.

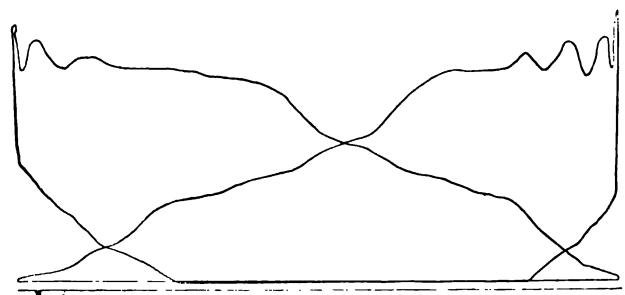

FIG. 289.—Indicator cards taken before removing the load and tracing the spiral shown in Fig. 288.

Let us now consider the so-called inertia effect. The point I wish to bring out is this: that if we add an inertia weight, or an inertia wheel, in the way that it is represented on page 940 of this paper, we may retard, instead of accelerate, the action of the governor. Mr. Mansfield has stated that for a large fly-wheel it has very little accelerating effect, whereas it may be readily demonstrated that if wrongly applied it will have a

retarding action, so that a governor with the inertia wheel would require a much longer time to move from one position of adjustment to another than it would with the wheel removed.

No experiments were made on the inertia weight, although we intend to do so, but the action of the inertia weight may be readily solved in an approximate way, in which all the errors involved will be against the point I wish to prove, which is that an inertia weight may be put on so as to retard the action of the governor.

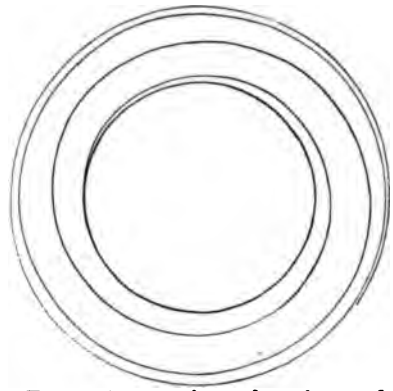

FIG. 290.—Tension of springs adjusted so that governor is in stable equilibrium. No dash-pot. Revolutions 245 per minute.

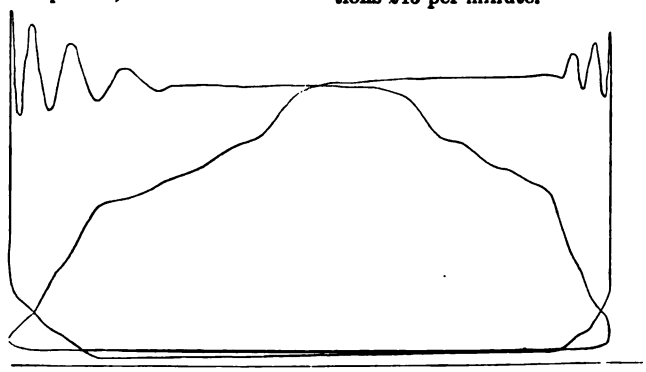

FIG. 291.—Indicator cards taken before removing the load and tracing the spiral shown in Fig. 290.

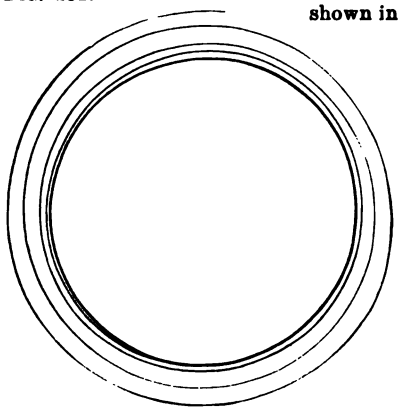

FIG. 292.—Adjustment of governor same as in Fig. 290, with less variation of load than that corresponding to Fig. 290. See indicator cards Figs. 291 and 293.

Let us consider the case of a loose fly-wheel, as represented on page 940 of the paper. Assume that the loose fly-wheel had 60° angular movement forward, which conforms with cases of actual construction, and determine the amount of variation in speed that would have to take place if the loose fly-wheel moved through its extreme range

in one, two, three, and four revolutions. If moved through the entire range in one revolution, the speed of the fly-wheel would have to vary $33\frac{1}{3}\%$ in order that the governor should move through the extreme of its range of 60°. This is on the assumption that the fly-wheel is uniformly accelerated, in which case we have $\frac{2 \times 60}{360} = 33\frac{1}{3}\%$. If moved through the extreme range in two

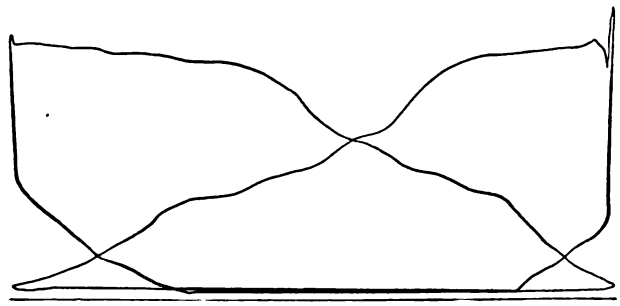

FIG. 293.—Indicator cards taken before removing the load and tracing the spiral shown in Fig. 292.

revolutions, the speed would have to vary $33\frac{1}{3} \div 2 = 17\%$. In three revolutions it would have to vary 11%, and in four revolutions 8%.

Taking the case of the Ball & Wood engine, in which these diagrams were obtained, we can readily determine how much an inertia wheel, placed loosely on the shaft, would move backwards for each revolution if the load driven by the engine is suddenly removed. Assuming that the whole effect of the steam acts on

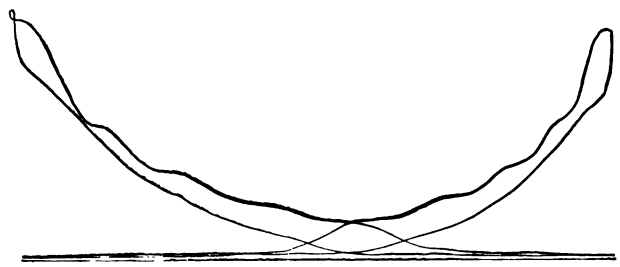

FIG. 294.—Average indicator cards taken after removing the load from the engine and tracing the spirals represented in the figures. Scale of all cards = 40 lbs. per inch.

the engine after the load is removed, we may determine the minimum time required to move the inertia wheel through a given

angle, or the maximum angle that the inertia wheel will move through per revolution of the fly-wheel shaft.

If the inertia wheel were used in the ordinary way the resistance of the valve would tend to increase the time required to effect an adjustment. The point of cut-off would also be earlier at each successive stroke after the load was thrown off, which will also increase the time of action, so that it would be greater than the minimum time determined by assuming the entire effort of the steam to act on the fly-wheel as already described.

In the case of the Ball & Wood engine there are two fly-wheels, each 5 feet in diameter, and their combined weight is equivalent to about 1600 lbs. at the rim. The cylinder is 10 × 11 inches, and the engine is rated at 50 H.P., at 300 revolutions per minute. If, while running at 50 H.P., the load is suddenly removed, the gain in speed per revolution, as calculated by Rankine's formula, and assuming no diminution in the indicator cards, is 1.8%. The movement of a loose fly-wheel for this variation of speed would be $0.018 \times 360 \div 2 = 3.2°$. For two revolutions the movement would be $3.2 \times 4 = 13°$. For three revolutions the movement would be $3.2 \times 9 = 29°$; and for four revolutions, $3.2 \times 16 = 51°$. Therefore, if the entire power of the engine were employed to accelerate the fly-wheel, the amount of movement of the loose wheel would be 51° in four revolutions.

We have not, as already stated, considered the effect of the action of the valve mechanism in retarding the velocity of the loose wheel, or of the diminution of acceleration of the fly-wheel on account of the reduction of work in the cylinder at each successive revolution after the load is suddenly removed. Both of these effects would tend to retard the quickness of action of the loose wheel so that a less angle would be moved through than is indicated by the calculations.

For three and one-half revolutions the maximum angle that the loose wheel could move through is $3.2 \times 3.5 \times 3.5 = 39°$. The experiments demonstrated that the governor moved over the extreme range in three and one-half revolutions, so that if an inertia wheel having an angular movement of over 39° were used in connection with the governor it would retard instead of accelerate its action.

I do not wish to be understood as speaking against the use of the so-called inertia governor, because if rightly applied it may hasten the adjustment, and combined with the dash-pot to prevent

overtravel it may be made to combine quickness of action with stability and close adjustment of speed. If a dash-pot is placed on a governor which has but little power, it retards the action considerably. An inertia weight having a small angular advance and producing a powerful action of the governor will produce the best results, and this should be borne in mind in designing a governor.

Mr. Frank H. Ball.—I had not intended to say anything on this paper, but a thought occurred to me since the paper has been presented that I wish to speak about. I think there is a tendency on the part of many of our members to make a distinction between inertia and centrifugal force that the circumstances hardly warrant. Of course, if we stop to think of it, we all know very well that centrifugal force is nothing but one manifestation of inertia. Inertia is that property of matter which resists any change of condition as to motion, and in the shaft governor we have centrifugal inertia and tangential inertia. We often see governors with one weight suspended so as to be acted on by tangential inertia and called an "inertia weight," and another arm from the same pivot carrying a weight located so as to be acted on by centrifugal inertia, and called a "centrifugal weight."

These weights move as one mass, and the forces which act at right angles to each other may be resolved into a single force acting at some intermediate point where the combined masses of the two weights may be considered as concentrated, and it therefore becomes merely a question of the angle of suspension of a single weight.

Mr. C. W. Baker.—Mr. Mansfield has told us that it is possible to design a governor which will regulate the speed without any appreciable change of velocity between full load and friction load. It would be of interest to the members if he would explain just what he means by appreciable change, because it is customary in drawing specifications nowadays to put in a limit as to what change in speed will be allowable; and there is some difference of opinion as to what is a reasonable requirement.

Prof. J. Burkitt Webb.—There are two or three points in the discussion that I would like to call attention to. First, with reference to something Professor Jacobus spoke of. If we attach a spring to a support and hang a weight upon it, it does not matter whether we pull the weight down three inches or six, and release it, it will take the same time to go back to the neutral position.

It is therefore quite in line to believe, as Professor Jacobus states, that whether we throw half the load or all the load off, the governor will take the same time to adjust; again, if we alter the sensitiveness of the governor by altering the adjustment of the spring, it will make no difference in the strength of the spring, and therefore the same thing will follow.

Coming now to the action of centrifugal force upon the spring in the governor, to which Professor Lanza referred, there is such an action, but it has not much effect, as I believe, on the result.

A spiral spring in a fly-wheel is usually in the plane of the wheel, *i.e.*, at right angles to the shaft, and generally does not lie in a radius of the wheel. It will therefore be acted on by centrifugal force in two ways: First, it will be bowed outward from the centre of the wheel as if it were (or, as it really is) a beam supported by its two ends. Second, it will have its pitch altered, so that the pitch will be greatest at that point where a perpendicular from the centre of the wheel falls upon the axis of the spring, and so that the pitch will decrease as we recede from that point.

Now, as to the effect of bending, it is easily seen that by changing somewhat the position of the fixed end of the spring we can allow for the bending, and so make the movable end meet the piece to which it is attached at the angle desired. This adjustment may be disturbed, but only slightly, by the spring assuming different lengths as the governor changes position.

As to the effect of the change of pitch, we may say, in general, that the application of a force or forces in the same direction as the spring, and to any point or points thereof, simply changes the pitch without altering the strength (number of pounds required to change length of spring a certain amount). Now, as the action of the governor depends on the strength of the spring, its action will not be affected by the change of pitch, although this, as well as the bending, may need to be allowed for in adjusting it for the proper speed.

There is one other point that Mr. Ball spoke of. First of all, however, I must object strongly to the improper use of the word "weight" in this discussion. It is not weight that we put in a fly-wheel governor; it is *mass*. We do not want the weight there, and, for the reason that gravity puts the weight there in spite of us, we have to balance it out. So we ought to speak of *mass*. Mr. Ball suggested that in governors using two masses on

each side of the centre, we might find some one mass to take the place of the two. That is not necessarily the case, for the reason that a mass cannot be put all strictly at one point, and consequently, if we have a mass attached to an arm, the arm and mass constitute together an extended mass of a certain shape, and two masses attached to the same pivoted piece constitute with that piece nothing more than a single mass of a certain shape. In other words, any mass that we use cannot be condensed into a point, but must be extended over a certain space and have a shape, and it is simply a question of what shape is the best. In calculating the centrifugal moment in a governor, this shape must be taken into account if the calculation is to be correct; in fact, in some governors, as the Gruson and Buss governors, the shape (or, otherwise stated, the fact of there being two masses) is the important element on which these governors depend for their peculiar actions.

Professor Lanza.—I will say that I do not think Professor Webb's reply has answered my question at all. Certainly when the springs bend in consequence of centrifugal force, some effect will be produced on the action of the governor, and I asked Mr. Mansfield whether he had taken account of this effect either by calculation or experiment.

Professor Webb.—The lengthening of the spring will not alter the strength of it. It will not alter the time of the vibration.

Mr. F. M. Rites.—In general, and making all allowances for approximation, the criticisms of Professor Jacobus on inertia governing are correct, except that he has not gone far enough. Under the best conditions of governor adjustment, the whole power of the engines is not utilized for acceleration until adjustment is complete; but, instead of that, only an average of the difference of the extremes of load and the time required to produce the required deflection by inertia alone would be roughly double what he has allowed.

It can easily be appreciated that an inertia weight may act as a brake and obstruction to the governor, and instead of being an active element in the adjustment, consume a great part of the centrifugal force and delay final adjustment. In such a case, a multiplication of inertia force only increases the evil, and introduces a new objectionable result; for after adjustment is complete, the entire governor system has acquired its maximum velocity relative to the carrier, so that the cut-off continues vary-

ing far beyond the requirements, and long after acceleration has ceased to be a moving force and becomes a negative one.

Improved results with such an inertia weight proves not that it was desirable so much as a redesign of the governor proper. Under test the inertia force is deceptive; for, while permitting closer regulation, it frequently, as has been pointed out, delays action by becoming unexpectedly negative.

These remarks, however, do not apply to inertia governors so called because of the addition of inertia force to that of centrifugal in the flying weights. In such a combination the governor is always steadier, permits closer and quicker regulation, and in every respect is better for the union of forces.

It must not be assumed, however, that centrifugal weights are necessarily evils, but it is a fact that the writer knows of no application of the inertia weight where the action could not be improved by an omission of the weight coupled with an intelligent redesign of the governor proper.

It is believed, however, that there is much to be discovered in shaft governor forces, and it is the intention of the writer to present to the Society at the first opportunity new facts and forces of great interest to engine builders.

Mr. Mansfield.—I am asked what I would consider an appreciable change in velocity. It is common in specifications for engines for electrical purposes to insert a clause requiring that the extreme variation of speed be not greater than two per cent. You all know that there is a great variety of designs of shaft governors, so much so that it has seemed, from what little experience I have had, that it is hard to make a shaft governor that may not be adjusted to hold an engine within two per cent. variation. The question might be asked, What is meant by a two per cent. variation? Is an instantaneous variation referred to, or a variation determined by counting the number of revolutions per minute? I believe if the number of revolutions by any means is counted, per minute, or per half minute, or per quarter minute, the variation may be practically imperceptible; that is, if the shaft governor is as well designed as may be, the only variations are practically instantaneous. I believe that reliable experiment has shown that they may be designed, especially when dash-pots are added to them, so that the variation may be in the unusual direction of increasing speed with increasing load. Therefore, we may adjust short of that point.

Mr. John A. Brashear.—I would like to offer this Society, if the Society does not have it in its possession, the scientific memoir of Leon Foucault, who made a most exhaustive study upon governors and their action, in his work which was largely devoted to the study of optical surfaces. The work was published by his wife, and I have an extra copy. I do not know that there is any particular virtue on my part in presenting it, but if Professor Hutton hasn't it in the library I should be glad to give it to the Society.

Professor Hutton.—It is not there, sir.

Mr. Brashear.—It is magnificently illustrated—a quarto of some 700 pages.

The President.—The book will be a very acceptable addition to the library.

DXCV.*

NOTES ON THE CORROSION OF A CAST STEEL PROPELLER BLADE.

BY FRANK B. KING, SPARROW'S POINT, MD.

(Member of the Society.)

It has been suggested that a note on the corrosion of the propeller of the steamer *Alabama* may be of interest to the Society, or may elicit information in the possession of others upon the curious phenomenon of propeller corrosion.

The *Alabama* is a fast passenger steamer plying in the waters of the Chesapeake Bay. The propeller in question was 12 feet 9 inches diameter and 19 feet pitch, and had four cast steel blades mounted on a cast-iron boss. The normal revolutions are about 98 to 100, making the speed of tips about 4,000 feet per minute.

The corrosion was peculiar in its rapidity and its distribution. Instead of being confined, as usual, to the back of the blade, well out toward the tip and leading edge, it covered about two-thirds of the back of the blade; in fact, only a small area toward the centre and along the following edge escaped. The working face was in good condition. Such was the rapidity of this corrosion that, after running four months, it was found advisable to renew the blades in cast-iron.

An analysis of the drillings taken from the four blades and one spare blade exhibited the following characteristics:

	Carbon.	Silicon.	Phosphorus.	Manganese.	Sulphur.
1...	0.35	0 30	0.038	0.58	0.046
2.....	0.31	0.22	0.034	0.51	0.042
3..... ..	0.35	0.25	0.032	0.55	0.046
4.....	0.35	0.34	0.030	0.57	0.040
5.....	0.46	0.30	0.030	0.65	0.034

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

DISCUSSION.

Mr. H. B. Roelker.—I found just that condition in the earlier steel blades imported from England (Naylor Vickers & Co., Sheffield) about the year 1872–3. They corroded away so quickly in salt water that the engineers said "they melted like sugar." The importation was then abandoned, and it was quite a long time before steel blades were made here which were sufficiently satisfactory to induce a regular business. The corrosion was always first and greatest on the back of the blades near the outer front end, but I have also seen considerable of it appear on the working surface, and I have seen holes five to seven inches diameter eaten through the blades in less than six months. I do not know the composition of the steel.

Mr. Gustavus C. Henning.—I think that corrosion of propellers in salt water (and very much more rapid in brackish water) is due to the air, which follows in the apparent vacuum back of the blade, which is abstracted out of the water surrounding that vacuum. The position of the corrosion depends on the precise shape of the leading edge of the blade and the outer edge and hub. If that is in such a position as to keep the vacuum right back of the blade following in toward the hub, the corrosion will be great. If the water is thrown around in a rapid eddy the corrosion will occur near the outside edge. I think it is due solely to the exhausting of air out of the water into the apparent vacuum back of the propeller. I think that the first propeller blades corroded so rapidly that they did not know what composition to use to prevent corrosion. We know that a pure steel corrodes rapidly. They furnished a steel that was too pure, and it corroded rapidly. Now the steel used in propeller blades answers well; it does not corrode as rapidly as it did with the superior materials that were used originally.

Professor Hutton.—What are the elements of impurity that diminish the liability to corrosion?

Mr. Henning.—I cannot say off-hand, because it is a combination of things. It depends on the metallurgical process as well as the composition. I will try to find it, and put it on record in connection with the discussion.

The President.—We had a similar case where a concern built a

mine pump for us, and, being ultra-conscientious, they put into it the same iron they were putting into machine tools. The result was that when the mine water got hold of it, it made very short work of it, and then we put in the hardest kind of iron we could.

Mr. Henning.—As a rule, the higher the carbon the less the corrosion.

The President.—In this case it would seem that the open pores allowed the water to get in, and the iron was very much more attacked.

Mr. Henning.—That is just as it should be, because in the gray iron the carbon is partly free, not in chemical combination, while in the white iron the carbon is absorbed in the metal; and the gray iron has pores which will allow the water to get in there, and hence the gray should corrode faster than the white iron.

The President.—The result was that all the parts that were exposed to water we rebuilt ourselves out of what we supposed to be very inferior material, but they are running yet.

Mr. J. J. Flather.—It suggests itself to me that the remarks of Mr. Henning last night in regard to the corrosion of bolts having different voltaic compositions would, in part, perhaps, account for this corrosion. The earlier propeller blades may have been less homogeneous than those of to-day. It is well known, I believe, and we saw the fact in the experiments reported by Mr. Henning last night, that there is a galvanic action set up between the different particles in the same iron, and it may be that the acid of the sea-water set up a galvanic action between the electro-positive and the electro-negative particles of steel in the same plate, and that this corrosion was due to that galvanic action. Did I understand Mr. Henning to say just now that the action of the air on the back part of the plate would make a greater corrosion there?

Mr. G. C. Henning.—Yes, sir. If you have no air the corrosion will not occur so rapidly; if you partly immerse, or immerse and remove from the liquid which contains salt or acid, corrosion will be rapid. If you keep it in pure water without any air the corrosion would be very slow. The galvanic effect, with the air, would cause rapid corrosion on the back of the blade, but without the air there the whole blade would corrode on account of the galvanic action.

Mr. Flather.—The British Admiralty experiments showed the

same thing, and they now require that zinc be connected by intimate contact with the plates on all marine boilers to prevent the corrosion which would otherwise take place.

Mr. H. B. Roelker.—I think I never saw any difference in the corrosion of propellers, whether working at or a little above the surface of the water, or far below. The propellers which work the quickest and hardest show the greatest corrosion every time.

DXCVI. *

THE RELATION OF THE DRAWING OFFICE TO THE SHOP IN MANUFACTURING.

BY A. W. ROBINSON, SOUTH MILWAUKEE, WIS.
(Member of the Society.)

THE purpose of this paper is to describe the system employed by the writer in the drawing office of his company in the hope that some of the points may be of use to members of the Society.

The drawing office is the origin of thought and action for the entire works as far as the design and construction of its product are concerned. It is responsible for the accuracy of its drawings and orders, and its authority should be unquestioned and above reproach in the shop. The shopmen should habitually trust and adhere to their drawings, and their faith should not prove to be misplaced. To maintain this there must be unceasing care and vigilance on the part of the office, and full adaptation to the shop needs and capabilities. It goes without saying that every drawing office, whether employing one draftsman or a hundred, should have its system and methods adapted to the needs of the establishment with which it is connected. As these needs vary with each case, it is not to be supposed that the system about to be described will be of universal application. It will be well then to state in a general way the conditions which this system is intended to meet. We will assume, therefore, that the office employs from ten to fifteen skilled draftsmen, and is in connection with a manufacturing establishment, doing a general engineering business in which there is comparatively little duplication of orders, and in which single orders frequently involve a large amount of detail, of which it is essential to keep exact records. It is also assumed that the drawing office is invested with the sole right and authority to

* Presented at the Montreal meeting, (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

issue orders to the shop for all new work, or all work in which there are changes and variations from previous similar work.

The practice of issuing verbal orders or directions for the conduct of work is productive of misunderstanding and confusion. When no evidence of authority exists no responsibility can be fixed. It is therefore advisable to have a system of written orders to all departments whereby the duty of those concerned is clearly defined, and the responsibility can be fixed for dereliction of duty.

SHOP ORDERS.—An order being once entered on the books of the company, the procedure is as follows: The business office issues a written order to both the drawing office and the shop upon a blank which merely states the general name of the machine, the time of delivery promised, and the number of specifications to be worked to if any, and the number by which the order is to be known. It is the duty of the drawing office to prepare such specifications beforehand when necessary. On the receipt of these orders in the shop, if it be a repair or duplicate of something already made, so that the shop superintendent has the information by which to execute it, he does so. If, however, it is new, or in any sense special work, he cannot proceed until the orders come down from the drawing office.

The drawing office issues orders upon the pattern shop and foundry by means of blanks headed "foundry" or "pattern shop," as the case may be, arranged thus:

B. S. S. & D. CO. ENGINEERING DEPARTMENT.

FOUNDRY ORDER.

ORDER NO.	DATE.	DRAFTSMAN.
	189	

Countersigned by	Examined by

These are manifolded in triplicate, and can be made out by any draftsman to whom the job is delegated, but must be signed by the chief engineer, or, in his absence, the chief draftsman. The two copies are then sent down to the shop superintendent's office, who keeps one on file for his own reference and information, and immediately sends the other to the foreman of the department for which it is intended. In this way the shop superintendent retains control of his men in the different departments, and has knowledge of the orders that are issued. He alone is responsible for their proper execution, and undue interference of the draftsman with the foremen or workmen is obviated.

It is also the duty of the drawing office to order all raw material for new and special work, that is not regularly kept in stock. This is done by blank as follows:

B. S. S. & D. CO. ENGINEERING DEPARTMENT.

DATE.	Please order the following
189	for Order No.

Ship

Draftsman	Countersigned,
	Chief Engineer.

These are simply requisitions on the business office, and the copy goes to the storekeeper as a statement that the articles noted have been this day ordered. He will therefore be expecting them, and on their receipt will at once know for what order they are intended. His copy of the manifold reads "The following material has this day been ordered for order No. —.

Written orders are not issued from the drawing office to any other departments, except the pattern shop and foundry. Drawings and sketch sheets are issued to the other departments, as

machine, smith, and erecting shops, etc. These pass through the hands of the shop superintendent, and in themselves constitute an order to make what they represent or call for, provided they are covered by the original general order from the business office and bear the same order number.

By means of these written orders to each department, each foreman knows definitely what work he has on hand, and all responsibility for errors or delinquencies is at once traceable to the culprit.

The shop superintendent is also empowered to issue written orders to his foremen in all departments, for all work which does not require information from or the authority of the drawing office. For these he uses his own blanks—those of the drawing office being labelled "Engineering Department."

It frequently happens that on large orders involving much detail, it is desirable to push the construction of parts as fast as they are determined upon before the completion of the general design. In these days of urgency and high pressure this is almost a necessary evil, but should be pursued with caution in the drawing office, lest difficulty be found in fitting the later part of the design to the earlier. Under this system of written orders the parts can be ordered in as fast as they are ready, even though the shop superintendent is as yet uninformed as to the balance of it, and does not know of what the complete order is to consist.

When the drawing office work on the order is completed an order list is made out and typewritten in duplicate. The order list enumerates in detail all the items making up the complete order, and is divided up into headings such as (1) castings, (2) forgings, (3) miscellaneous, (4) special material ordered outside, and so on. For each item is given a reference number of the drawing or sketch sheet on which it is shown, and it is or should be shown thereon so fully and definitely that no further questions need be asked. This order list is essential: 1st, to inform the shop definitely of what the work consists; 2d, to refer the shop to a source of information concerning each and every item; 3rd, to form a shipping list so that in shipment nothing will be overlooked that should be sent; 4th, to form a permanent record by which repairs may be readily identified, and from which future machines may be compiled or adapted. These order lists are press copied in a book for the purpose.

DRAWINGS.—The primary function of a shop drawing is to answer the shopman's questions, and indeed it may be said that this is its only function. There are certain things connected with the material, form, dimensions, finishing, fitting and erecting of a machine and each part of it that each department needs to know. Sometimes in simple cases the various processes of pattern-making, finishing and erecting can be defined on one drawing, and in other cases separate drawings containing separate information for the different processes are demanded. It is important to have all the necessary information conveyed on a drawing in a direct and legible manner, and that the views be so chosen as to represent the object in the simplest way. Let the draftsman, on beginning to make a shop drawing, say to himself, "Now what does the fellow who is to make this want to know?" and then let him put down just that information and no more, but be sure and get it all on. Refrain from all superfluous lines and marks, and make the drawings so plain that "he who runs may read them."

The mere ability to make lines and circles and projections is really the least important and least valuable part of a draftsman's skill. Neatness and accuracy of drawing is desirable, but if it is obtained with the expenditure of an undue amount of time, and does not carry with it a practical knowledge of shop needs and shop processes, it ceases to be a virtue.

The following set of rules for the drawing office have been found to be useful and to work well. They contain some points that a good draftsman ought to know, but they are incorporated as reminders, and as being necessary to preserve uniformity of practice among changing draftsmen.

DRAWING OFFICE RULES.

Shop Drawings.

1. All drawings shall be of the uniform size of 23″ × 36″.

2. All detail drawings for use in the shop shall consist of whole standard sheets, half standard sheets and sketch sheets. Half-sheets shall be 18″ × 23″, formed by ruling a line across the centre of standard size sheets as filed, the blue prints only to be cut, and mounted and varnished, when necessary.

3. The sketch sheets shall be 8″ × 11″, and shall be used for

all simple details, forgings, for bolt lists, and for all temporary work capable of being shown in this way. All standard machines shall be fully drawn out and blue printed. The sketch sheets shall be made with indelible pencil or copying ink and press copied in the book for the purpose. The information on the sketch sheets shall be as complete as that specified for drawings.

Character of Shop Drawings.

4. A shop drawing is to be considered as an order or instructions to the shop, and not merely as a statement or illustration. For this purpose it must convey clearly all the information needed to make and finish the article.

5. Every dimension necessary to the execution of the work is to be clearly stated by figures on the drawing, so that no measurement need be taken in the shop by scale. All measurements to be given with reference to the base or starting-point from which the work should be laid out. In comparatively simple constructions the several parts are to be shown together complete, although each part must be figured independently, and details supplied, if necessary, by sketch sheet. In more complicated forms each part should be detailed by itself and a general drawing made showing the thing complete. No details should be sent out without putting them together on a drawing, or taking them from a general drawing, so as to insure their fitness. Unnecessary duplication of views to be avoided, except in display or advertising drawings.

6. All figured dimensions on drawings to be plain, round vertical figures, not less than $\frac{1}{8}$ inch high, and formed by a line of uniform width and sufficiently heavy to insure printing well. No thin, sloping, or doubtful figures or diagonal-barred fractions will be tolerated. All figured dimensions below 3 feet to be expressed in inches.

7. All centre lines to be alternate dot and dash in fine black line. All dimension lines to be in continuous red lines, with a central space for the figure, and of such strength as to show on blue print more faintly than lines of drawing. Lines of drawing to be bold and clearly defined in proportion to the scale, and to be shade-lined by making the right-hand and bottom lines heavier. No ornamental shading or other "frills" allowed on shop drawings.

Title.

8. Every drawing, whether whole or half-sheet, shall have the title, date, scale, and number of the sheet placed in lower right-hand corner. One man will be detailed for this duty, to secure uniformity.

9. The name of the drawing, as given in the title, is invariably to consist of two divisions in one line separated by a hyphen. The first division is to state the general name of the thing or machine, and the second division is to clearly designate the part or parts represented (or if a general view, should so state). The wording of the titles should be submitted to the chief engineer or head-draftsman for approval.

Drawing Symbols.

10. Detail shop drawings should state:

(1) The pattern number of every casting, in plain figures of larger size than the dimension figures.

(2) The number of each piece required for one set. This should be written in one word (not figures) and followed by symbol of material.

(3) The material of which the parts so ordered are made, using symbols as follows:

C. I. — Cast Iron.	Bs. — Brass.
W. I. — Wrought Iron.	Bz. — Bronze.
C. S. — Cast Steel.	Bbt. — Babbitt.
M. S. — Machinery Steel.	V. F. — Vulcanized Fibre.
H. S. — Hammered Steel.	C. R. S. — Cold Rolled Steel.
Other materials write full name.	

(4) The kind of finish on each of the different parts will be indicated by a letter preceding the figured dimensions, as follows:

F.—Means "Finish," and indicates that the surfaces to which it applies are to be machined or dressed in suitable manner to size stated.

F. B.—Means "Finished Bright," or polished.

G. F.—Means "Grinding Finish," and indicates that the only finish to be allowed is that necessary for grinding.

When no letter precedes the figured dimension it is under-

stood that the part is to be left black or rough. In cases where finish might be presumed but not required, follow the figured dimension by the word "Cast" if a casting, and "Rough" if a forging.

(5) A reference list of sketch sheets that may be used for detail illustrations.

Sketch Books.

11. Each draftsman will be supplied with a sketch-book by the company, and in which he shall make all his notes, calculations and data referring to his work, and under no circumstances shall original work be done on loose sheets and transcribed into these books. No effort should be made at neatness or nicety in these books, but each entry should invariably be commenced with the name of the thing and the date, and full notes made of data on which the calculations were based, and the results obtained clearly stated. These books are to be the property of the company.

Index for Drawings.

12. An index book for drawings will be kept in the drawing office by the clerk. This book will be divided into as many divisions as there are drawers, with provision for indexing 100 drawings in each drawer. The names of the drawings will be added to the various divisions according to their classification. The system of numbering shall be as follows:

13. Each drawer shall be numbered consecutively, and shall contain drawings devoted to a certain class of work, which shall be indicated on the drawer label. The drawing number shall consist of two or more digits with a decimal point between them. The whole number shall indicate the number of the drawer, and the figures after the decimal point shall indicate the serial number of the drawing in that drawer. For example: Drawing No. 5.16 is the 16th sheet in Drawer No. 5, and Drawing No. 75.96 is the 96th sheet in Drawer No. 75. Not more than ninety-nine drawings shall be put in one drawer, except in exceptional cases.

14. Sketch sheets will bear the number of the letter and page of the letter book, preceded by the letter S, to distinguish them from drawings, and will be indexed in their own impression

book, but not in the drawing index book. They will be referred to on general drawing of which they are details, and will also bear the number of such drawing.

15. When making a new drawing the draftsman will apply to the clerk for a number, and will be allotted the first unappropriated number in the division to which the drawing will belong.

16. On completion of every drawing or sketch sheet it must be examined and initialed by the engineer before being issued, and the following entries made in books kept for the purpose:

(A) Record of blue prints and sketch sheets issued to shop, giving date, number, and title.

(B) Drawing index.—Record in daybook the number, title, and sub-title, draftsman and date.

(C) Pattern index.—Record in daybook the number, classification and correct name of patterns, with remarks and date.

Each draftsman will see that these entries are properly made.

Patterns.

17. All patterns shall be numbered with the number of the drawing from which they are first made, followed by a letter indicating its serial on that drawing. For example, if four patterns are shown in detail on drawing No. 36.50, the patterns shall be numbered 36.50A, 36.50B, 36.50C, 36.50D. When existing patterns are utilized in a new design or machine, their original number is to be noted on the drawing on which they are shown in their new employment.

18. Upon receiving formal notice from the pattern shop that patterns are ready for inspection, the draftsman connected with the order shall examine same and issue foundry order for the casting. The date of inspection and name of inspector shall be entered upon the pattern maker's report at the time of making such inspection.

The sketch sheets referred to in these rules are 8 x 11 inches in size. They are of stiff cardboard, and the heading is printed in copying ink. The sketch is made with an aniline copying pencil, the "Eagle No. 2" in wood being used. They are press-copied in books for the purpose, and several books are used for

different classifications of work. The books are of slightly heavier tissue paper than is commonly used for correspondence, and have 500 leaves each, numbered consecutively throughout the series, so that the number of a sketch sheet is never duplicated. In this system there are six books for copying sketch sheets, representing as many classes of work, and these divisions will readily suggest themselves as required for any particular case.

The use of these sketch sheets is especially for work which does not require to be often duplicated, and for giving quick despatch to emergency work. A free-hand sketch can be made, copied, and issued in this way in ten minutes, while the regular process of drawing, tracing, blue printing, and waiting for the latter to dry, or the sun to shine, may consume hours. They have the additional advantage of being more convenient to handle and file away in the shop than blue prints, and they save multiplication of tracings and consequent drawer space. The copies being in book form cannot be lost and are easily indexed and consulted. Sketch sheets are convenient for rapid detailing of forging and small parts, and such parts need, therefore, to be merely indicated on the general or erecting plan, and reference numbers given of the sketch sheets. The sketch sheets will also all appear as items in the order list.

In addition to the stiff card sketch sheet, it is convenient to have a "Drawing Office Memorandum" blank. This is a copying-ink heading printed on a sheet of letter paper, and is used for order lists and all sketch matter sent abroad from the drawing office.

After copying, the sketch sheet, if for permanent use, is sized with a mucilage, composed of gum tragacanth and water, and then varnished with white shellac and alcohol. The sizing is to keep the lines of the sketch from running while varnishing.

Two necessary articles of office furniture are the drawing table and the blue-print frame. So many excellent forms of these have been devised that it seems hardly necessary to refer to them in this connection, but sometime hereafter occasion will be taken to describe types which have the merit of cheapness and effectiveness.

It is our practice not to finish original drawings, but to trace from them on tracing cloth. These tracings are used only to print from and are filed away in a fire-proof vault. Two prints

are made of each tracing as soon as finished, one for the shop (or more if necessary) and one to file away in the drawers of the office. These drawers are 24 inches × 38 inches × 2 inches, and are each calculated to hold a maximum of 100 prints. In this way the tracings are preserved from risk of fire and loss and from the wear of frequent handling.

As a rule each draftsman makes his own tracings, and only skilled draftsmen are employed. The writer does not advocate the employment of cheap draftsmen to trace shop drawings from the originals of the designer. If this is done the designer must finish his original to entire completeness before turning it over to the tracer, thus consuming additional time and running more risk of errors and omissions than if he traced it himself. A skillful draftsman will merely block out his entire work on the original and give his whole thought to the perfection of his design. In the tracing he can re-arrange his drawing if necessary, and the time occupied in tracing is usually much less than that employed in working out and perfecting the design, and a draftsman worth $120 per month, will usually trace twice as fast as one worth $60, and do it better.

The titles on drawings are mainly done by rubber stamps giving the name of the company, the number of the drawing, and having spaces for the insertion of name, date and scale.

Some experimenting was done to find a suitable stamp ink for tracing cloth. Printers' ink was tried, but it rubs off and does not dry satisfactorily. A special lithographic ink is used, which is similar to printers' ink, but with addition of a dryer. It is applied to the stamp by a composition roller in similar manner to printers' ink and gives a black impression which blue-prints well. The number stamp has movable type.

As a rule it pays to employ only high-class labor in the drawing office. A draftsman puts his own impress on his work, his individuality goes into it, even if closely supervised, and it is upon the perfection of detail that the success or failure of a new design mainly depends; it is important that the draftsman entrusted with it shall have the necessary skill and ability.

We have appliances for testing the efficiency of almost every known mechanism, but who can measure the efficiency of a draftsman? We appreciate the economic value of good steam distribution and the like, but too often is the efficiency of the draftsman neglected, and thousands of dollars spent in the construction

of work which would have yielded much better results if a little more brains had been used in its design.

In conclusion, the writer would say: Do not have so much system that it is difficult to work to or burdensome to carry out. A few simple rules, faithfully adhered to, are better than the most elaborate system which is loosely or imperfectly carried out. The object of a system is to define the duties of each man and to fix the responsibility of dereliction of duty.

DISCUSSION.

Mr. L. S. Randolph.—There is one point in which I cannot altogether agree with Mr. Robinson, and that is with regard to the size of sheets. I have found smaller sheets so much more useful than the large, especially for detail work, except where the parts detailed were very large.

I have found a very convenient system was to make the largest sheet 24 × 36 inches, and divide this up by halving the larger dimensions; this makes a series as follows: 24 × 36 inches, 18 × 24 inches, 12 × 18 inches, 9 × 12 inches, 6 × 9 inches, 4½ × 6 inches. This system is in use in several drawing offices. The objections to the large sheets are principally the increased cost of both labor and material. They are more inconvenient to handle in the shops, and more likely to be torn or broken, this being particularly the case when mounted and varnished. When the drawing is not very complicated it is really harder to read a large than a small drawing.

Mr. Robinson's method of sketch sheets is the most promising I have ever seen, and should, if well handled, do away with one of the frequent complaints against drawing offices—of preventing rapid execution of work by a cumbersome system.

Mr. Louis S. Wright.—This paper is especially interesting to me, and, I think, would be to many others who have been in any way connected with organizing or managing a manufacturing establishment with a large drawing office. I wish to say that, in the main, I agree with what Mr. Robinson has said. There are some detailed arrangements, however, with which I cannot agree, probably owing to the fact that the needs of Mr. Robinson have been somewhat different from those with which I have been connected.

With reference to size and shape of drawings, Mr. Robinson

advises all shop drawings to be made of one standard size, namely, 23 × 36 inches. I feel that there are but few cases where this could be carried out, and then it would be only where the product of manufacture was some standard article, where there would be time to re-draw the details of design to get them within the compass of a single-sized drawing paper. To do this in an establishment where the work was not duplicated would cause unallowable delays, and would prevent your using the original drawing of whatever size it might be to send to the shop. In every case, I should certainly advise adhering to some standard size for drawings, but not confining it to one particular size of sheet.

With respect to materials ordered for manufacturing the product and designs of the drawing office, to my mind it is certainly wisdom that these orders originate in the drawing office, as they are more likely to be accurately specified both in quality and quantity. The material can also be ordered at an earlier date than if time is given for others in the establishment to order it. In connection with this it would seem to me advisable that all orders for material, of no matter what character, pass through the hands of a man in charge of stores, so that nothing is ordered which is unnecessary. This will oftentimes use up quantities of material which would otherwise lie in store for an indefinite time.

With respect to sketch-books for the individual draughtsman, I consider this a great saving of time and money, as no calculations have to be made a second time if the sketch-books are used religiously. We use a sketch-book of a similar kind, 6 × 10 inches, and with about two hundred pages of yellow paper of such a finish as to allow using pen and ink, the paper cross-ruled in quarter-inch squares, making it convenient for either writing or making sketches. These books, as the draughtsman may leave the employ of the company, revert to the drawing office clerk, who keeps them in charge for reference, and we have found it to be a most valuable system.

With respect to the relation of the drawing office to the pattern shop, we consider that our pattern shop cannot be too close to our drawing office; in fact, we will probably build a new pattern shop adjoining the drawing office, and bridge across from one to the other, so that draughtsmen can have easy access to the store of patterns. This will oftentimes force them to use patterns by making a slight alteration in them, which otherwise would be overlooked, occasioning the expense of new patterns. We con-

sider it essential that the pattern shop be under the direction of the drawing office, and have our drawing office issue written orders to the pattern shop for each pattern that is to be made, the pattern shop reporting when the pattern is completed; the pattern shop foreman, together with the draughtsman who examines the pattern, signing the written order certifying to its correctness.

With respect to numbering patterns, we have found it most desirable to number all patterns consecutively, placing raised letters on the patterns, so that the letters showed clearly on the casting at all times. Cases frequently come up with castings far from the works where this proves to be invaluable.

Mr. Robinson strongly recommends that the draughtsmen trace all their own drawings, and that none but high-class men be employed. The writer does not agree with him in this, and has found a cheaper plan to be to employ high-priced leading draughtsmen to do the thinking and make the design, employing cheaper men or boys to trace these designs, holding certain leading men responsible for the correctness of the work, and doing this in the following manner: Giving to each of the leading men once a year a hundred dollars, provided no errors causing expense have been made. Each leading man so responsible signs the drawing examined by himself, and, in the event of errors, the cost of correction is calculated at once by the superintendent in charge of the shops, and recorded on a blank form especially for the purpose. One-fifth of this cost is deducted from the hundred dollars going to the leading man, and if at the end of the year he does not earn anything it is very evident a good plan would be to get rid of him.

This regulation has worked admirably, and, we consider, has saved us many times the amount of money paid in premiums to the leading men, the moral effect of which is good.

Mr. Henry Binsse.—There is one point in the paper which I think is extremely unpractical. He speaks of putting all dimensions under three feet in inches. So far as my knowledge goes, the measuring tool of the machinist is a two-foot, not a three-foot rule. The yard is not used by machine builders. The practice recommended by the paper would compel the machinist to do sums in addition and subtraction, and would lead to errors which might easily be avoided by making the standard of measures recorded on the drawings compare to the standard carried by the machinist, which is the 24-inch rule.

Mr. A. J. Shaw.—We have in use in our works a pattern numbering system which has been found very convenient for our particular business, the building of cranes and hoisting machinery. All patterns are divided into classes, corresponding to the usual names, and the classes are designated by numbers. For instance, all truck brackets go into class 1, all trolley sides into class 2, and so on throughout the list. There are in all some forty classes, and, so far, we have not had occasion to make a pattern which could not readily be referred to some one of them. The patterns in each class are numbered consecutively from one up.

Each pattern is designated by the class and serial numbers written together in fractional form, the class number for the numerator, and the consecutive number in the class for the denominator. For instance, the twentieth truck bracket will be written as $\frac{1}{20}$, and spoken of as "one over twenty."

This system, of course, does not refer the pattern back to the drawing on which it is shown. That we do not consider necessary, as it is very easily found out from the pattern list.

The particular advantage of this system is, that on account of all similar patterns being grouped together, a draughtsman by going to the pattern list can find at once all the patterns of that particular kind which we have on hand.

In the pattern loft we divide the patterns in just the same way, all of each class being kept together and, as far as possible, arranged on the shelves in numerical order. This makes it possible for any one familiar with the system to go directly to any pattern wanted. If, for example, a draughtsman wishes to see what bracket patterns are on hand, he can go to the pattern loft and find all of the patterns of that kind, which we have grouped by themselves, instead of having to hunt through a miscellaneous lot of material, as has been customary where the patterns are numbered consecutively throughout, or from the machine on which used, or the drawing on which shown.

We have found that this system results in a great saving of time, and also prevents much needless duplication of patterns, which is likely to obtain under other systems, and some four years' use has developed no disadvantages.

Mr. Thos. W. Capen.—The paper so nearly agrees with my own practice, that I find little in it to criticise and much to approve. The points of difference between the author's practice and my own are these: I have not found it desirable to

have the draughtsmen write orders, but have employed a special clerk, who was sufficiently acquainted with drawings to be able to read them with facility. After the tracings were approved by the chief of the department, they were turned over to this clerk, who indexed them, had them blue-printed, and then wrote his orders on the purchasing agent for outside material, on the pattern shop for castings, and on the smith shop for forgings. These orders were all copied in a press-book, and extra copies also made for the shop superintendent, thus notifying him what material to look for for the job. The foundry and smith shop returned these orders with the weights of pieces added, and they were kept on file to assist in estimating. Draughtsmen were not allowed to inspect patterns, this being made the duty of the foreman pattern maker, the writer believing it better to leave this duty to one man than to many. The writer differs from the author again in the execution of the drawings, and has found a great saving in time in omitting all shade lines, and has experienced no difficulty in having the drawings readily understood by the various mechanics who work from them. Light dot and dash black lines are also preferred to red for centre lines.

On the whole the writer regards the paper as a valuable contribution to the literature of the subject, and hopes that it will bring out much valuable criticism, as there are many drawing-rooms sorely in need of such a system.

The President.—As it is so close to the time of adjournment, I shall adjourn the meeting. But I would like to add one word, and that is, I think if most drawing-rooms would spend a few hours or a few minutes in making a large blue print to be placed in the business office with these words on it: "Do not take any orders which agree to deliver yesterday things ordered to-day," there would not be so much confusion in the drafting department.

*Mr. A. W. Robinson.**—The foregoing discussion brings out the fact that differences in practice must exist to meet different needs, and it is interesting and instructive to compare them.

The "special clerk" employed by Mr. Capen to read drawings, order raw material from them, as well as issue orders in detail for the shop-work, and notify the superintendent of his actions, must be a man of versatility and skill, whom none but the largest con-

* Author's Closure, under the Rules.

cerns can afford to employ. As a rule this work needs the most careful oversight.

Mr. Binsse's objection to thirty-six as the numerical limit for inches is simply a matter of practice or preference. Some shops use sixty inches as a limit, and above that the inches certainly become cumbersome. A man accustomed to measuring does not need to calculate, and everybody knows that thirty inches is two feet and a half, and it is a simple expression, and takes up less room on the drawing. Boiler plates are always listed in inches.

I commend Mr. Shaw's plan of grouping similar patterns in the pattern storehouse by classes. They need not necessarily be grouped by class number. For instance, gear patterns of certain pitches, sprocket wheels for certain chain, sheaves, bearings, and the like, are all stored in groups by themselves, as they are liable to be used in any class of machine.

Mr. Randolph's objection to size of sheet as being too large is a matter of preference and requirement of business. Much of our work is so large that it cannot be shown on a small sheet, except to a microscopic scale. The sketch sheets take the place of the small sheets described by Mr. Randolph, so that few large sheets are needed, and they are simply assembling or general views. I have found it desirable to use the sketch sheets liberally in order to restrict the number of drawings required, and thus save much drawer space and general cost of drawing. The size I recommend as standard (twenty-three by thirty-six) is a stock size of a superior linen ledger paper which we buy by the ream from paper stock dealers in flat sheets, "true to gauge," at one-tenth the cost of the various brands sold in rolls by the draftsmen's supply houses.

It will be seen (with reference to Mr. Wright's discussion) that although I recommend but one standard size of sheets, and thus have only one size of drawer to keep them in, I really have three sizes to issue to the shop, inasmuch as half standard sheets are used, and the eight by eleven sketch sheets. The half sheets are never cut in originals or tracings as filed, but only as blueprints.

DXCVII.*

FIRST STATIONARY STEAM ENGINES IN AMERICA.

BY F. R. HUTTON, NEW YORK CITY.
(Member of the Society.)

A LETTER was recently addressed to the Secretary of the American Society of Mechanical Engineers, asking the following questions:

"What, when, and where was the first steam engine in America (not locomotive and not steamboat engine)?"

Reflection brought out the point that this was a question on which the *Transactions* of that Society might well contain information, and that, as far as known, the historical text-books were silent, or at best indefinite, upon a matter which has at least a historic interest, and concerning which much information might doubtless have been transmitted verbally, and would be in the possession of persons now living. It would obviously be more difficult to secure this information as time went on.

The locomotive engine in this country has been quite thoroughly written up in the memorials of the Stevens family, in the papers of Mr. Horatio Allen, and most fully of all in the exhibits in the Transportation Building of the Columbian Fair in Chicago, 1893.

The history of navigation by water has again been very thoroughly reviewed in the lives of Robert Fulton; in pamphlets detailing the early Stevens experiments, and in the History of the Growth of the Steam Engine by Prof. R. H. Thurston, member of this Society. While there is a regrettable silence and lack of record as to the development of engineering for water propulsion during the period of its most active growth, from the beginning of the century up to the period of the War of the Rebellion, yet the first beginnings are matters of pretty full record.

James Watt died in 1819. The period of his greatest activity

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

runs from 1774, when the partnership with Boulton was begun, up to the first year of the present century, when the two partners turned the business over to their sons and retired from active business. There could not have been any well-known constructor of steam engines in the neighborhood of New York City in 1807, when Fulton prepared his *Clermont*, or he would not have sent to Boulton & Watt, in England, for his cylinder.

In Pennsylvania, Oliver Evans, in 1800 or 1801, built an engine "six inches diameter of cylinder and eighteen inches of stroke of piston, which he applied with perfect success to driving a plaster mill." * Oddly enough, he called this engine the "Columbian" engine.

It then occurred to the writer to make individual inquiry among those conversant with such matters, with most interesting results, which it is the purpose of the remainder of this paper to record, in the hope that the contributors may be induced in the discussion of it to give even fuller detail, and that by this publication other engineers may be induced to take part in filling up the record and rendering it complete.

Mr. W. F. Durfee writes:

"In the last year of the eighteenth century there were but three steam engines in the United States. One of these was used (probably built as well) by Oliver Evans in Philadelphia for grinding plaster; another was an imported engine, built by Hornblower and brought to America by his son, and put at work for draining the Schuyler copper mine in New Jersey. The third engine is said to have had a locus in New England, but just where I have never been able to discover."

The first engine referred to above is the same one mentioned by Prof. Thurston. The second one was for a mine at Belleville, near Newark, and all that appears to remain of the engine is its cylinder, which was imported from England in 1753. The boiler is said to have been of copper, upright, 8 or 10 feet in height, and of equal diameter. It will be remembered that this cylinder was an object of interested inspection on the occasion of a visit of this Society to shops of Watts-Campbell & Company, in Newark, N. J., December, 1886. It was not bored, but was cast surprisingly round, and used as it came from the foundry.

* History of the Growth of the Steam Engine. R. H. Thurston, 1886, page 155.

Mr. Samuel Webber writes: "I find in looking up my old memoranda, that the 'second steam engine in Providence, R. I., of 24 H.P., was built by Oliver Evans, for the Providence Bleaching, Dyeing and Calendering Co., in 1814.'

"The next date I have is a steam cotton mill at (Olneyville) Providence, in 1831, but whether this was the old 'Providence Steam Mill,' which was one of the first, if not quite the first one, I am not now sure. Your correspondent must look to some Rhode Island authority to settle this point. I have also a note of a steam mill in Fall River in 1831, but cannot say what one; their first mills were all 'water-power.' North of Boston the first one was the 'Bartlett Steam Mill,' of Newburyport, in 1838, followed by 'James,' in 1843, then came the 'Ocean,' of the same place, in 1846, and the Portsmouth of same date."

Believing that in a manufacturing centre like New England the textile industry would be among the first to call for the steam engine, in pursuance to the letter of Col. Webber, inquiry was made at the hands of Mr. C. J. H. Woodbury, and from his letter in reply the following quotation is made:

"I also wrote to the venerable Thomas J. Hill of Providence, a man nearly ninety years old, and still in active business in the manufacture of cotton machinery; and he gave a reply which confirmed my impression, but gives many other details. He says that the first cotton mill run by steam that he knew of was the Steam Cotton Mill of Providence, R. I., built by Samuel Slater, Wilkinson, Howe & Tompkins of Pawtucket, and Benjamin and Charles Dyer of Providence, during the years 1828 and 1829, and was started under the supervision of David Whitman with 4,300 spindles. They made at first sheeting with number 38 warp and number 45 filling, the warp being sea island cotton costing sixty-eight cents a pound, and the filling upland cotton costing nineteen cents a pound. The mill changed ownership on account of dull times in 1829, and Samuel Slater took it with Trott & Bumstead & Fessenden of Boston, as partners. This mill was not successful at first, and changed superintendents quite a number of times and made but little money until 1838."

Mr. Chas. W. Copeland, in reply to an inquiry, writes a letter full of most interesting details, from which the following quotations are taken:

"My father built the first engine for power purposes in, I

think about 1826 or 1828. My father [Daniel Copeland] had the largest machine shop in Connecticut, and the only one in that State which built steam engines and boilers; there was a shop in Boston by a man, I think named Ashcraft, who built small engines; I remember that he had an application to build a 20-H. P. engine, and he declined because it was too large for him, but he would build for him two engines of 10-H. P. each, to work together.

"When my father took his first engine to build, it was for a blowing engine for a foundry in Hartford, and my father went to New York to get men to go to Hartford to work on it as there were no men in Hartford who had any experience in that class of work. I worked on the engine when building, also when being erected; I do not know what afterward became of it.

"About the time of which I am writing, a Mr. James (the same who put the first locomotive on the Harlem Railroad) built a small engine for a coffee and spice factory; it had two cylinders, in fact a compound engine, only the cylinders had both the same bore and one cylinder exhausted into the other; Mr. James claimed that this tended to economy of fuel—I differed from him, and I remember we had quite an argument."

As the history comes up to more modern days, the uncertainty of course diminishes, and history becomes more full and complete. Mr. Washington Jones, of Philadelphia, consulting authorities under his hand, gives the following references:

From an abstract of a lecture delivered at the Franklin Institute, Nov. 20, 1885, on "Oliver Evans and his inventions," by Coleman Sellers, Jr. "In 1803, Mr. B. H. Latrobe, in his report to the American Philosophic Society, describes five or six engines then at work in the United States, and among others mentioned a small engine erected by Mr. Oliver Evans. This was doubtless his first engine, that which he started the year previous to Mr. Latrobe's report."

Mr. Jones, also quoting from a historical sketch of Oliver Evans, by Rev. Geo. A. Latimer, adds that "the cost of the Evans' engine of 1801 was $2,000, and that during the next year he became the first regular steam-engine builder in the city of Philadelphia, at the Mars works, corner Ninth and Vine streets."

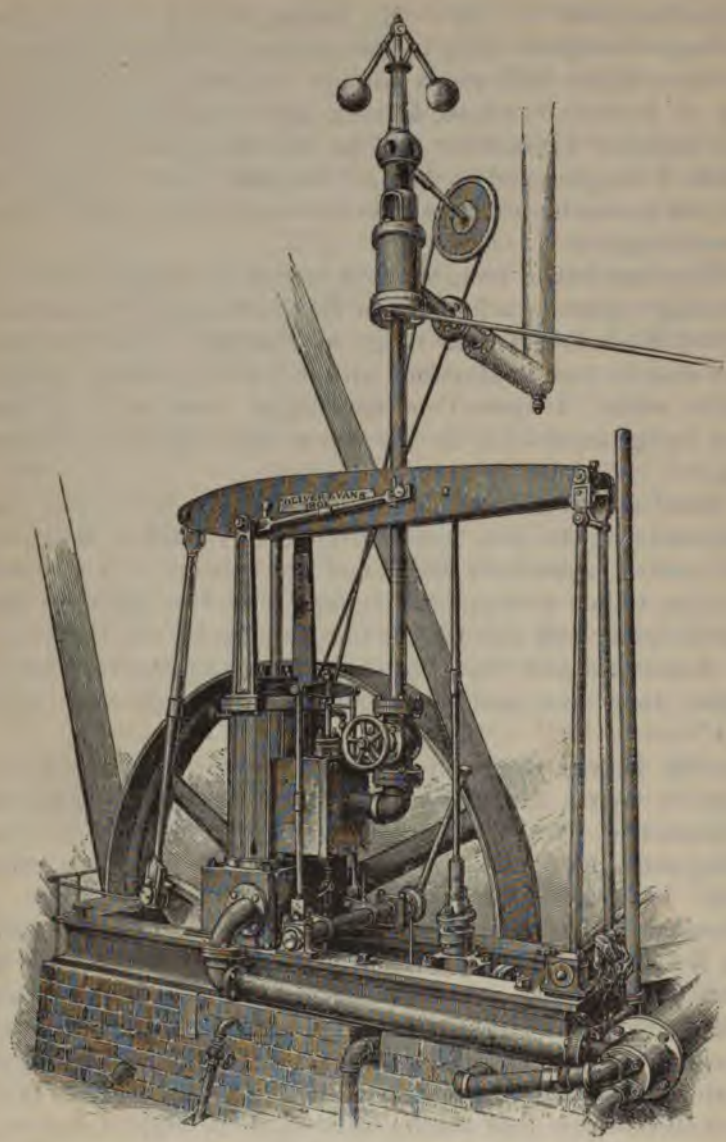

FIG. 295.

Another Evans' engine was sent to New Orleans for a steamer, but "financial difficulties and low water combined to prevent the completion of the steamer, and the engine was set at work driving a sawmill, where, until the mill was destroyed by

fire, it sawed lumber at the rate of 250 feet of boards per hour." *

Mr. Jones also quotes from a history of Philadelphia by Scharf and Westcott, vol. i., page 517: "Manufactures are growing rapidly; of calico printing establishments near Philadelphia there were Howson's at Kensington, Stewart's at Germantown, and Thornburn's at Darby. The three turning out 200,000 yards in 1803." Mr. Jones adds, "Two at least of the above-named must have been driven by steam-engines."

Other Evans' designs in Philadelphia were the "Grasshopper" engine at the Burtis cotton-spinning mill in Philadelphia, whose design stamps it as being either of Oliver Evans' make or of Rush & Muhlenburg, his sons-in-law, partners and successors. The mill building was closed and the engine dismantled about 1866.

Mr. Jones also met a non-condensing engine in the distillery of Alexander Young's Sons in Philadelphia, which was built by Hyde & Flint, of that city, and had been in constant use for sixty years. It is of the overhead beam type, steam cylinder, 12 by 36, with a long slide valve.

The general appearance of these Evans' engines is made apparent from the cut herewith (Fig. 295) reproduced with its permission from the "Iron Age," Jan. 25, 1894. It was built by the I. P. Morris Company, of Philadelphia, in 1845, the date upon the beam being that of the Evans' patent. It remained in constant service in the smith shop of the above company until 1893, when it was presented to the University of Pennsylvania. The only repair necessary was the re-boring of the cylinder from its original 8½ to 10 inches, and the patch put on the upright of the parallel motion, which was done by Mr. Jones himself. The stroke is 18 inches.

A similar inquiry also brought from Professor Sweet the suggestion that at the Brush Electric Light and Power Company of Savannah, information could be had concerning one of the very old Boulton & Watt engines which was still running in that city. Fig. 296 is reproduced from a blue print furnished by Mr. T. P. Keck, of that company, showing the engine as restored by Messrs. John Rourke & Son, of Savannah.

The legend on the plate gives full detail.

These records are made public to the members of the Society,

* Thurston, page 158.

and have been compiled both for the sake of what they themselves contain, and more especially to elicit either from members themselves or from those whose interest they can secure, facts which shall supplement, confirm, or controvert those which this note has aimed to record.

DISCUSSION.

Mr. W. F. Durfee.—I am in receipt of your very interesting paper on "The First Stationary Steam Engines in America." Possibly the mill in Fall River which Mr. Webber states was driven in 1831 by steam was the American Print Works. I am sure these had a steam engine before 1840.

In a paper by the late Fred. Graff, entitled "The History of the Steam Engine in America," published in the *Journal of the Franklin Institute* for October, 1876, from which I quoted largely in the discussion of Allan Sterling's paper on water-tube boilers (*see Transactions American Society of Mechanical Engineers*, etc., vol. vi., pp. 588, 589), there is an account of the engine erected at Centre Square, Philadelphia (where the new Public Buildings are now), for supplying the city with water. This engine was put in operation in 1801, and was abandoned in 1815. The cylinder of this engine was bored in Paterson, N. J. (I think), and several weeks were required for the work. I will suggest that you print an illustration of this engine, which was one of the earliest built in America. I have a strong suspicion that the restoration by Messrs. John Rourke & Son of the Boulton & Watt engine at Savannah was something more than restoration. Farey, *On the Steam Engine* (1827), does not give any representation of any such valve gear or arrangements of "side pipes" and "steam chests" or "condenser" as is shown in the plate. The eccentric rod is also suspiciously modern. The "beam," "connecting rod," "parallel motion," "cylinder," "air" and "feed" pumps are very likely Watt's, but the "valve gear," "steam chests," "side pipes," "eccentric rod," and "condenser" (that is, its position under the cylinder) are too exact a reproduction of those used on the Sound and the Hudson to have been made by Watt in 1815. Stuart, in his *Anecdotes of the Steam Engine* (1829), does not show any such perfection in valve gear. In an article communicated to the *Scientific American* (March 17, 1894) I give some account of "Avery's rotary

engine," of which a considerable number were made and sold by Messrs. Lyndes & Son, of Syracuse, N. Y., in 1832–35. The engine described was used to drive the presses of the *Railroad Journal*, and its editor, Mr. Minor, speaks in high terms of its "compactness, the ease with which it is managed by any person who can tend a fire, the trifling cost of fuel, as well as the small outlay for the engine;" and goes on to say that "its most important advantage will be found in its perfectly uniform motion." This engine ran at a velocity of 5,000 revolutions per minute. I have reason to believe that the early engine whose locus I could not fix when I wrote you, was actually used in New York City for driving a saw-mill. Perhaps some of the old ship-builders could tell you something about it. Trusting that the foregoing may be of service in the revision of your very interesting paper, I remain,

Yours truly, W. F. DURFEE.

P.S.—The Baldwin Locomotive Works commenced building stationary engines quite early. I think there is one still in operation in New Bedford which was built by them in 1836.

Mr. Henry Binsse.—On reading your short paper on the "First Steam Engine," I applied to Mr. Charles Bradley, of Newark, the

FIG. 297.

great-grandson of Josiah Hornblower, for more detailed information. With great courtesy he gave me two pamphlets on the subject, one entitled *Josiah Hornblower and the First Steam Engine in America*, by William Nelson, secretary of the New Jersey Historical Society; the other a pamphlet written by a member of the Hornblower family, describing the various steam engine

improvements made by them. He also gave an impression of the Hornblower family seal, showing the power-house of the original engine. No other drawing or picture of the engine exists. I enclose a copy of a letter by Justice Bradley, who made a careful examination of the subject, which was also lent to me by Mr. Bradley. I shall present the pamphlets to our library, and recommend them to those interested in engineering history.

There are a few facts in Mr. Nelson's paper which I shall add to Mr. Justice Bradley's letter.

The engine was ordered in 1748 or 1749, and Benjamin Franklin refers to it in a letter of February, 1750, saying that the cost of the engine was £1,000 sterling. This was evidently the price of the machinery in England, for, later on, a gentleman who visited the engine at work records that it had cost, set up, about £3,000 sterling. It is to be hoped that this note will not cast a feeling of gloom over the engine-builders present at the meeting. It is quite clear that they were all born about one hundred and fifty years too late, and that the earliest were also the golden years of the steam engine.

A little over four years later the engine was shipped from London, arriving, after a three months' voyage, in September of 1753. It might be suspected from this delay that the intention was to make Colonel Schuyler believe that he was about to receive very full value for his money; but, although there may be something in this, it is more likely that it was found very difficult to secure permission to ship the machine, as England, with all a mother's tender solicitude, expressly forbade any of her colonies from playing with machinery of any kind whatever. No machinery was allowed to be exported. The exception in this case was probably due to Colonel Schuyler's high position. A year and a half was spent in erecting the machine, pump, power-house, and whatever was required to start the engine. The duty of the engine was about eight hogsheads of water per minute.

Josiah Hornblower, the first steam or mechanical engineer in America, became a very prominent man, going to the New Jersey legislature, afterwards to Congress, and filling many highly respected positions. He was the father of Chief-Justice Hornblower, and his descendants have made their mark in many lines of life.

There was another steam engine erected in New York City in 1774, for a water-works, by Christopher Collis, which you can add to your engine list.

[MR. JUSTICE BRADLEY'S LETTER.]

WASHINGTON, *September* 29, 1875.

DAVID M. MEEKER, Esq.

Dear Sir—The steam-engine, of which you possess a relic, was, as you suppose, the first ever erected on this continent. It was imported from England in the year 1753, by Col. John Schuyler, for the purpose of pumping water from his copper mine, opposite Belleville, near Newark, N. J. The mine was rich in ore, but had been worked as deep as hand and horse power could clear it of water. Colonel Schuyler having heard of the success with which steam-engines (then called fire-engines) were used in the mines of Cornwall, determined to have one in his mine. He accordingly requested his London correspondent to procure an engine, and to send out with it an engineer capable of putting it up and operating it. This was done in the year named, and Josiah Hornblower, a young man then in his twenty-fifth year, was sent out to superintend it. The voyage was a long and perilous one, and Mr. Hornblower expected to return as soon as the engine was in successful operation. But the proprietor induced him to remain, and in the course of a couple of years he married Miss Kingsland, whose father owned a large plantation adjoining that of Colonel Schuyler.

The late Chief Justice Hornblower was the youngest of a large family of children which resulted from this marriage. Mr. Hornblower's father, whose name was Joseph, had been engaged in the business of constructing engines in Cornwall from their first introduction in the mines there, about 1840, and had been an engineer and engine-builder from the first use of steam-engines in the arts—about 1720.

The engines constructed by him and his son were the kind known as Newcomen engines, or Cornish engines. That brought to America by Josiah was of this description. Watt had not then invented his separate condenser, nor the use of high pressure; but it is generally conceded that, for pumping purposes, the Cornish engine has still no superior. After 1760 the Schuyler mine was worked for several years by Mr. Hornblower himself. The approach of the war in 1775 caused the

operations to cease. Work was resumed, however, in 1792, and was carried on for several years by successful parties. It finally ceased altogether early in the century, and the old engine was broken up and the materials disposed of. The boiler, a large copper cylinder, standing upright, 8 or 10 feet high, and as much in diameter, with a flat bottom and a dome-shaped top, was carried to Philadelphia. The relic in your possession was a portion of the cylinder, and was purchased by some person in Newark. In 1864 I met an old man named John Van Emberg, then one hundred years old, who had worked on the engine when in operation, in 1792. He described it very minutely, and, I do not doubt, accurately. It is from his description that I happened to know the kind of engine it was, although, from the date of its construction and the use to which it was put, there could have been but little doubt on the subject.

What changes have been wrought in one hundred and twenty-two years! What mighty power has been created on this continent in that time by the multiplication and improvement of the steam-engines! We may well look upon this relic with a sort of superstitious veneration, looking forward as well as backward, and wonder what another century will bring forth.

Respectfully, your obedient servant,

JOSEPH P. BRADLEY.

Mr. O. C. Woolson.—In the discussion of the first stationary steam-engine in America, the early history of municipal water-works occupies a prominent part. The New York *Times*, in an issue in September, 1874, printed the following, which is quoted without assuming any authority of the exactness of its statements:

"The first water-works in the United States appear to have been planned and constructed by Mr. J. C. Christensen, at Bethlehem, Pa., more than a century ago, namely, in 1762. The machinery consisted of three single-acting force pumps, four-inch calibre and eighteen-inch stroke, worked by a triple crank, and geared to the shaft of an undershot water-wheel eighteen feet in diameter and two feet clear in the buckets. The total head of water was two feet. On the water-wheel shaft was a wallower of thirty-three rounds, gearing into a spur-wheel of fifty-two cogs, attached to the crank. The three

piston-rods were attached each to a frame, or cross-head, working in grooves, to give them a parallel motion with the pump. The cross-head was of wood, as well as the parts containing the grooves as guides. The water was raised by this machinery to the height of 70 feet, and subsequently to 114. The first rising main was made of gum wood, as far as it was subject to great pressure, and the rest was of pitch pine. In 1786 leaden pipes were substituted; in 1813 they were changed for iron. These works were in operation as late as 1832."

In the sixth volume of the *Transactions* of this Society, in the discussion by Mr. W. F. Durfee on "Shell and Water Tube Boilers," he makes very full and interesting reference to the old boilers of the Centre Square Water Works of Philadelphia. It is interesting to reproduce here the three illustrations given by Mr. Durfee, on pages 588 and 589 of that volume, in order that, in connection therewith, there may be presented some additional facts not included in Mr. Durfee's discussion. These are taken from a tracing copied from an original, which had the signature of Mr. Frederick Graff.

"TRACING OF STEAM BOILER IN USE AT CENTRE SQUARE WATER WORKS, PHILADELPHIA, FROM JULY, 1801, UNTIL AUGUST, 1812.

"The boiler was a wooden box, nine feet square and fourteen feet long, made of five-inch white pine plank, securely bolted

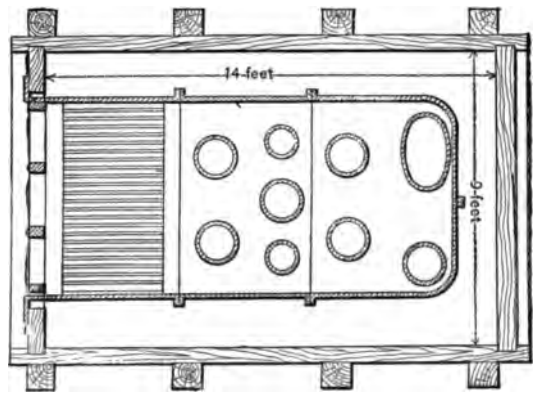

FIG. 298.

through, and braced with oak scantling on the outside (not shown in the drawing).

The first one made had a fire-box of wrought iron, with ver-

tical flue of cast iron. An almost similar boiler was afterward put in by Frederick Graff at Fairmount Water Works, except that a cast-iron case was substituted for the wooden one, and the flue and fire-box were of wrought iron. This latter boiler

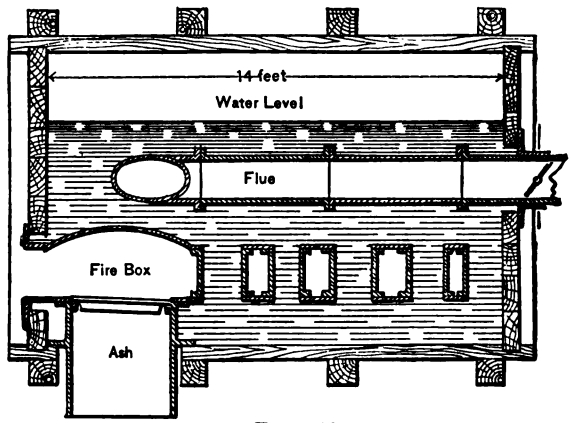

FIG. 299.

was in use from about September, 1812, to April, 1822. The Centre Square engine supplied from the above-mentioned

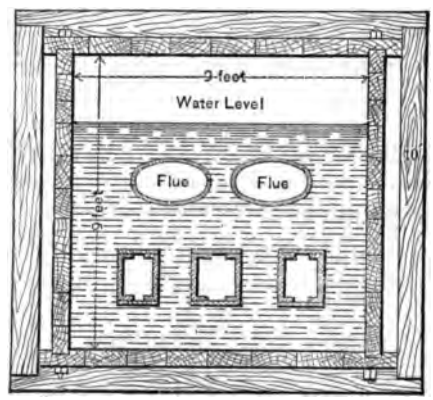

FIG. 300.

wooden boiler was 36 inches diameter and 72 inches stroke. It pumped, by experiment, 962,520 ale gallons of water in 24 hours, with 55 bushels of bituminous coal, and raised the water about 50 feet high.

"FREDERICK GRAFF,
"*Civil Engineer.*"

Mr. Thos. J. Borden.—Confirming Mr. Durfee's memoranda, the American Print Works was built in 1834, and was operated by steam power chiefly. While the Metacomet Mills were driven partly by steam, they were built in 1845, and so hardly belong to the very earliest period.

The Fall River Iron Works Co. built a large plant of rolling and nail mills in 1840–41, all operated by steam power, except a trip-hammer and a pair of shears which were driven by a water wheel. Their old works built in 1821 were operated partly by water power, but I am under the impression that they had some steam power nearly or quite back to their commencement.

Andrew Robeson & Sons began calico printing here about 1826, and probably used some steam power, but I do not know positively about that.

Mr. W. B. Le Van.—In regard to the first steam engine in America I would state that a condensing engine was built in this city by Thomas Holloway in 1819 for Francis Perot, and did good work for fifty-three years, and was then in perfect working order, but was only replaced as the Malting Company of the above-named party required additional power. It is now an object of curiosity in the courtyard of the malt-house. This engine was drawn through the streets of Philadelphia in the Constitutional parade held in 1887, when a great deal of interest was manifested in it.

*Mr. F. R. Hutton.**—I am much obliged to the gentlemen who have taken the trouble to supplement and confirm the record I had sought to establish. I hope the closure of this discussion will not mark the dropping of the subject by the engineers of America, where we have been so busy making history for engineering that there have been few to write it. With the death of an older generation will be lost much of oral tradition in these matters, and the loss will never be able to be made good.

In furtherance of Mr. Durfee's inquiry about the restoration of the Savannah engine, Messrs. John Rourke & Son reply to my letter as follows:

DEAR SIR:—Your favor of May 25th to hand, and contents noted. In reply will say that the side pipes, valve gear, eccentric rod, and steam chests, also the location of the condenser below the cylinder and supporting the same, are all as when the engine was erected in 1815. The only change has been a new piston and

* Author's Closure, under the Rules.

governor in place of the one shown on the blue print. The steam pressure has also been raised from 18 to 60 lbs. We have had the only drawing that was ever made struck off [traced?], and you have one of the blue prints. We remain

Yours respectfully,

JOHN ROURKE & SON.

I have also received from Mr. Thomas H. Walton, engineer for the Savannah Steam Rice Mill Co., in which the old engine is located, a letter under date of June 25, from which I make the following extract, as tending to render even more complete the detail of the record concerning this monument. I quote as follows:

"Mr. Rourke made no alterations in this engine. He fitted new brasses to the principal bearings and renewed those of the links and radius rods of the parallel motion. The steam nozzles and main valve chests are like those of an engine used to hoist coal, working at Washington Colliery, Durham, England, with the exception of a slip joint which is lacking in this engine in this mill.

"The condenser under the cylinder is the original one. On account of the unequal expansion of the steam nozzles when steam is first turned on, with the engine valves closed, the lower joint between valve chamber and steam way on cylinder has been broken. But if care is taken to heat up cylinder as quickly as possible after the steam is newly formed, this strain, caused by unequal expansion, is avoided. This proves that this engine was built before the use of the expansion joint was applied to this part of the steam-engine.

"The eccentric rod and the cam rocking shaft, as now used, are the original ones; and this goes to show that this system of operating valves was known when this engine was built. This feature struck me forcibly, for I always thought that this valve gear was first used on American steamboats. It may be important for investigators to know this much.

"The only important change made in this engine is the substitution of a Judson governor for the old-fashioned one, which is shown in the drawing marked Fig. 296.

"It is possible that some genius has been tampering with the main radius rods, because they are now set to bring the links at

one-half stroke in the center line of the cylinder continued, instead of their being in line at three-quarter stroke, as they should be, so that the upper ends of the links are as much on the inside of said center line on the top and bottom of the stroke, as they are on the outside of it when the beam stands horizontally, therefore the path of the cross-head deviates from center line.

"I cannot think that this mistake was made when the engine was first erected, nor can I accuse Mr. Rourke's men of the blunder; perhaps it is of older date than were the repairs he made a few years ago. No one knew better how to set up parallel motion than its projector.

"I am, my dear professor, very truly yours,

"THOS. H. WALTON,
"*Mining Engineer.*

"P. S.—I forgot to state that several years ago the old hemp-packing piston of this engine was changed and made to pack with rings and springs. The old follower is on hand. At the same time, as a matter of course, the cylinder was turned by boring. The old force pump wore out and was replaced by another of the same pattern. T. H. W."

In further confirmation of the purity of the design, it may be added that the historic Watt engines of this period, or before, preserved in the South Kensington Museum, in London, England, are also of this same general arrangement, even where the sun-and-planet motion is shown in place of the crank, and thus mark the engine as belonging to the period when there was a patent upon the crank.

DXCVIII.*

RUSTLESS COATINGS FOR IRON AND STEEL.

TINNING AND ENAMELING METALS, LACQUERING, AND OTHER PRESERVATIVE METHODS.

BY M. P. WOOD, NEW YORK CITY.

(Member of the Society.)

THE protection of iron and steel used in engineering, structural and art work, from the corrosive action of the atmosphere, as well as from air charged with the fumes of many industrial and chemical pursuits, and the fact that the usual coating with paint proves so ineffectual to guard from rapid decay, even when periodically renewed, has long attracted the attention of inventors and experimenters, as well as the best efforts of many chemists and scientists.

The latter, with all the resources of well-equipped laboratories, have not been much, if any, in advance of the less favored experimenter who developes his ideas by main strength and awkwardness, though, the fact or process once brought to light, the work of the savant is acceptable in proving the *rationale* of the same.

Frederic S. Barff, of Kilburn, county of Middlesex, England, and George and A. S. Bower (father and son), of St. Neets, county of Huntington, England, were the first to practically coat wrought iron, steel, and cast iron with a magnetic oxide, forming at the cost of the metal itself the protective layer, which is usually formed from paint. This is the process known as the Bower-Barff, that was developed in England about the year 1875.†

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* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Vol. XV. of the *Transactions*.

† See the United States patents—To Frederic S. Barff, No. 182,148, September 12, 1876, Protecting iron surfaces. Described in the *Official Gazette*, Vol. X., page 443. To George and A. S. Bower, No. 234,524, November 16, 1880, Coating iron with oxide. Described in the *Official Gazette*, Vol. XVIII., page 1147. No. 270,004, January 2, 1883, Depositing metals upon the surface of iron and steel coated with magnetic oxide. See *Official Gazette*, Vol. XXIII., page 440. No. 270,005, same parties, same date, for a furnace used in connection with the

The process, in brief, consists of placing the articles to be coated in a muffle, heated to approximately 1,000° Fahr., and when they have attained this heat, either by extraneous firing of the muffle, or by passing producer gas or the products of combustion through the muffle and amidst the articles contained, or by both methods named, steam superheated to the temperature of 1,000° Fahr. is then admitted to the muffle, and the formation of magnetic oxide begins at once. The heat of the muffle decomposes the steam, setting hydrogen free, which can be utilized to assist in maintaining the heat of the muffle; the oxygen uniting with the iron to form $Fe_3 O_4$, or magnetic oxide. The period of exposure is from five to twenty hours, according to the size of the articles and depth of coating required, which varies from $\frac{1}{100}$ to $\frac{1}{10}$ of an inch, and can be continued almost indefinitely, or until all moderate thicknesses of iron are resolved wholly into oxide. A very able presentation of the full process was made to this Society in a paper read by Mr. George W. Maynard, and found in *Transactions*, American Society Mechanical Engineers, 1884, Vol. IV. page 351. The color of the magnetic oxide so formed is a dull blue-black; the coating is tenacious, though when broken through by drilling, shearing, or mechanical injury, oxidation of the injured surface proceeds more rapidly than in untreated articles.

The high heat and intermittent character of the process is destructive to the retorts or muffles, and the process is necessarily prohibitory to all articles requiring cutting qualities. Its application to fire-arms instead of the customary bluing or browning by acid was not a success, too much attention being required to prevent the springing or distortion of the barrels, and removing the deposit formed in the bore, though it has been employed to some extent in the fittings of fire-arms. Articles distorted in process can be reheated to a low red or black heat and re-straightened without

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process. See also *Scientific American Supplement*, Vol. VII., Nos. 174 and 175. "History of the Bower-Barff Process," Vol. IX., No. 213, and Vol. XII., No. 287. For cuts and description of the furnaces employed in the process, Vol. XIX, No. 478. "Preservation of Iron and Steel by One of their own Oxides" (the Bower-Barff process practically considered), by B. H. Thwaite, Assoc. M. I. C. E., from selected papers of the Proceedings of the Institute of Civil Engineers, in which reference is made to a paper by Dr. Percy, F.R.S., Hon. M. I. C. E., "On the Protection from Atmospheric Action which is imparted to Metals by a Coating of Certain of their Oxides respectively." *Journal of the Iron and Steel Institute*, 1877, pp. 456, 460.

injury to the coating. The process materially weakens the iron or steel articles coated, not only by the amount of material changed from a metal to a metallic oxide or metalloid; but suffers a further loss of strength, particularly in wrought iron, by the burning out of the cinder in articles long exposed to the heat, making what rollers and forgers call "dry iron." A further difficulty appeared in structural work held in place by rivets; the heat necessary to form the oxide caused the loss of set in the rivet. If riveted together after oxidizing the parts, the rivet had no protection from corrosion, and Muntz or gun-metal connections became necessary. The coefficient of elasticity between the oxide and the metal from which it is formed, is so nearly the same, that railway bridge members and car truck frames are comparatively unaffected by long periods of service under compression, tension, and vibratory strains. The process applied to the pipes and iron work exposed to the action of mine, sulphurous or acidulated and sea water, appears to give that protection from rapid decay so long desired by those who "churn" those liquids. The data concerning the cost of application of the process are very unsatisfactory. So much depends upon the size of the articles, depth of coating, whether the furnace or muffle is operated continuously or not; but it must be an expensive process per pound or square foot on account of the skilled labor and superintendence required, setting aside the expense of superheating the steam required to the necessary temperature of 1,000° Fahr.

None of the parties using the process appear to be at all anxious to recommend the adoption of their steam superheating device, or to flit their tales of woe about the same. Costs given range from $5 to $20 per ton of articles treated, but there are many situations in which the protection given by this process would bear an addition to these figures. The amount of steam required if used alone for the oxidizing element and for coatings of a moderate thickness, is determined to be; for each pound avoirdupois gain in weight of iron oxidized to form the magnetic oxide, 232.9 cubic feet of hydrogen weighing 2 lbs. was set free, making 3 lbs. of steam or 75 cubic feet at or slightly above atmospheric pressure.

The above is the theoretical amount needed to furnish the oxygen; in practice from 25 to 33% more was passed through the superheater and muffle to get results. The deeper the coating the more steam was required. If 1,000 square feet of surface was

exposed for oxidation only $\frac{1}{64}$ inch thick, it equalled 625 lbs. of magnetic oxide, and would require at least 2,500 lbs. or 62,500 cubic feet of steam to be furnished in about six hours, or approximately 17 H.P. per hour. The patents for the process, if not expired by limitation or by the non-use requirements of foreign patent laws, are rendered of small value except in special cases by more recent improvements in the art by several inventors, one of which is a matter of record in this Society; and is of interest aside from its cheap and effective application by not being patented, the inventor by publication having given its use to the public.

The paper No. 180 presented to this Society by Wm. H. Weightman. Transactions A. S. M. E., 1886, Vol. 6, page 351, embraces a report by Lieut. Col. A. R. Buffington, Ordinance Dept., National Armory, Springfield, Mass., to the Chief of Ordnance, U. S. A., Washington, D. C., fully describing the process, which is peculiarly applicable to finished surfaces of iron and steel, notably fire-arms and kindred work; the process being conducted at so low a degree of heat as to not affect tempered parts of the arm.

Finished surfaces of cast iron are given a bronze-colored coating which sustains mechanical injury, such as a blow, without cracking the coating or affecting its protective power. The apparatus for armory work consists of a common cast-iron vessel or open-mouthed pot, deep enough to receive the longest piece to be treated (a gun-barrel immersed vertically) and large enough in diameter to contain a dozen or more barrels arranged on a movable rack capable of being raised and lowered into the bath of melted nitre, which contains a small percentage of peroxide of manganese. A brick furnace is built around the nitre pot, and a coal fire maintains the heat of the mixture, which is hot enough to ignite and maintain combustion in a pinch of sawdust scattered over the surface, or about 650° Fahr. The articles, polished but not burnished or unduly raised in lustre, are simply immersed and moved about in the bath for a given time or until the requisite color is obtained, which can be ascertained by raising the articles out of the bath and re-immersing them any number of times until the color is right; then they are removed, cooled in the open air to about the temperature of boiling water, then plunged into hot water to remove the nitre, and then into a sperm oil bath to complete the process.

The nitre used must be refined, the commercial article not giv-

ing uniform results. Articles not taking the desired color on the first application, or which bear a streaked or spotted appearance from some peculiarity in the metal, are corrected upon a second application. Care must be taken to prevent the introduction into the bath of water, oil, wood, or any carbon or combustible matter. The nitre mixture is not explosive, even when poured into the open fire, but in this case the arising fumes are suffocating and dangerous. The simple and inexpensive plant, the economy of time, labor, and material, the uniformity of product, and the beautiful color mark this process an important advance in the art. I have been unable to learn of the application of the process to any large unfinished members of architecture or other structural bodies, but I see no reason why it should not be as effective in these cases as it is in the hundreds of small articles comprising horse furniture and saddlery fittings, general hardware, and articles made of malleable iron, wrought iron, and drop forgings, which are on sale nearly everywhere.

The Journal of the Society of Arts has a translation by Mr. Frederic W. North, of a paper read before the Paris Société d'Encouragement, by M. Octave de Rochefort-Lucay, on the new Bertrand processes for coating with magnetic oxide and enamelling iron and iron carburets, and on a new process of tinning for cast iron, that are of interest as being possibly more simple and effective than the Bower-Barff process, and are evidently based upon a new discovery in chemistry which may be thus stated. If a thin adhesive film of another metal is formed on a steel, wrought or cast iron body, and this article be then heated to 1000° Fahr. and exposed to a current of oxidizing gas; the oxygen penetrates through the film, oxidizes the body metal, and forms magnetic oxide. The formation of this oxide continues indefinitely, the thickness increasing according to the period of exposure to the oxidizing current, provided the temperature remains at about 1000° Fahr. The film of metal deposited in the first instance disappears in some obscure way, forming oxides which mingle with the magnetic oxide or volatilize according to the nature of the metal of which they are composed. M. Bertrand's experiments to find the best metal for depositing the primary coating, led him to select bronze (a mixture of copper and tin) which gave perfect satisfaction, and for depositing it upon the articles to be coated with oxide, he uses electricity or wet baths, and sulpho-phenolic acid.

The following is the method adopted by M. Bertrand for an oxidation. The article, cleansed (which is indispensable), is dipped a few moments in a bath containing a solution of sulphophenate of copper andtin; a slight coating of bronze is formed; the article is immediately washed in cold water and dried with sawdust, then placed in the furnace. Oxide forms, and at the end of fifteen or thirty minutes (according to the size of the article) it is sufficiently oxidized, the coating formed varying from $\frac{1}{10}$ to $\frac{1}{8}$ of a millimetre for these exposures; longer periods give thicker coating to any desired extent.

M. Bertrand uses an electrical current to determine if the coating is of uniform and sufficient thickness. The wires leading from the poles of an ordinary galvanic battery are fitted with a bell arranged to ring when in circuit. If on application of the wires to the article being coated the bell rings, the current passes, showing the oxidation is insufficient; if the bell is silent, the oxide is of sufficient thickness to insulate the object and to practically protect it.

PROCESS FOR TINNING CAST IRON.

The coating of cast-iron surfaces with tin is one of the most unsatisfactory processes connected with the mechanical arts. M. Bertrand also uses sulpho-phenolic acid to obtain a durable coating of tin on cast iron. A bath is formed of water and the acid in which salts of tin is dissolved in the ratio of 1% of tin salt and 5% of acid. The article to be tinned, previously cleaned, is dipped into this mixture, and is at once covered with an adherent coating of tin, which can be afterwards polished on ordinary buffing wheels of wire and cloth and a result obtained which is at once cheap and effective.

PROCESS FOR ENAMELLING CAST IRON.

The records of the Patent Office show many formulæ for compositions, and designs of furnaces used in the process, for enamelling cast iron. But few if any of either the compounds or furnaces materialize on application, generally leaving the party to his own devices for accomplishing the desired results. There are only two processes by which a satisfactory enamelled surface can be given cast iron.

In the first, or hot process, the iron, heated to a bright red, is powdered by a sieve with a flux powder (the borosilicate of lead), then heated until the flux fuses, and is then powdered with a more fusible flux, forming the glaze of the enamel. This is the only

direct enamelling process, and is dangerous to the operator, expensive in application, difficult to apply to large articles, nor does it allow of decorations. The second process consists of coating the cast iron with a kind of pottery, by three distinct and successive operations in the furnace.

In the Bertrand process the article is first coated with the magnetic oxide as described above, then brushed over or dipped cold into the borosilicate of lead, colored with metallic oxides, to which is added a little pipe-clay to give the bath more body. The article thus covered is put into the furnace; the enamel adheres and vitrifies at the ordinary temperatures used by enamellers. By putting a coating of colored enamel with a brush on the first plain coat of enamel, it is possible to make any desired decoration, which may be burned in at one operation. The process is particularly adaptable for out-door vases, statuary, columns, railings, balusters, etc. The results due to the first oxidation with magnetic oxide are remarkable as much for their color as for the tenacity of the enamel and its resistance to rough usage. The process is adapted to articles of any size or shape, is not dangerous to apply, and is comparatively cheap in application. The back numbers of the *Scientific American* contain some valuable suggestions and information upon the above and other processes for the protection of iron, for which the writer cheerfully renders acknowledgment.

A process of forming a black or magnetic oxide that has a limited application, but worthy of mention as containing some possibilities for the future experimenter, is the Maritens Patent Process, in which the article to be oxidized is connected to the anode pole of a galvanic battery and immersed in hot water, when it becomes oxidized. A repetition of the process is necessary to secure a favorable result. Full information and detail of the process is given in a publication by Dr. George Langbein, of Leipzig, Germany, of which there is a translation with additions, by William T. Bramet, editor of the *Techno-Chemical Receipt Book*, 81 Liberty Street, New York. In this process the water is decomposed by electrolysis, the hydrogen being set free, while the oxygen goes to the iron to form an oxide coating. This process appears to be the progenitor of the electric welding process that attracted so much attention at the Columbian Exposition. In the latter case the strong current from the dynamo causes so rapid a decomposition of the water as to surround the immersed article with an envelope of hydrogen of so high tension that it prevents the escape

of the heat generated by the decomposition of the water, and a rapid rise in temperature to the welding point results; the oxygen being set free, no oxide forms. The weaker current used in the Maritens process simply reversed the welding method.

The first stage in the corrosion of iron is the formation of ferrous oxide, FeO, in which one atom of iron has united with one atom of oxygen (77.777% Fe, and 22.222% O). It is never found in nature in a free state, on account of the avidity with which it absorbs oxygen, and it is the active agent in the production of yellow ochre, that contaminates the waters in the mining regions.

The second stage, in which two atoms of iron unite with three atoms of oxygen to form Fe_2O_3, *red rust,* the chemical equivalents in this case being 70% iron, 30% oxygen; the magnetic or black oxide of iron, in which three atoms of iron unite with four atoms of oxygen, form an intermediary product, the chemical equivalents being

$$3\,(56) = 168, \text{ or } 72.41\% \text{ iron, Fe.}$$
$$4\,(16) = 64, \text{ or } 27.59\% \text{ oxygen, O.}$$

The oxide of iron found adhering to the plates of boilers under the water is usually the magnetic oxide. This black oxide is also produced, in some instances, when iron is placed underground in contact with wet cinders or sand. When this compound is formed it cannot easily be converted into the red oxide by action of air and water.

Iron heated to about 230° C. (446° Fahr.) combines directly with the oxygen of the air and forms a thin film of magnetic oxide of a yellow color which passes into red, blue, and gray. At a red heat this coat of oxide becomes thicker—*forge-scale*—and gradually passes at its outer surface into ferric oxide. At a white heat iron burns freely in the air with the production of magnetic oxide. At a temperature of 360° C. (680° Fahr.) iron decomposes watery vapor, forming in closed vessels magnetic oxide and liberating hydrogen.

Magnetic oxide is found native on the coast of New Zealand in the form of a black titaniferous sand, and, although constantly washed by waves of the sea, its stability is apparently unaffected.

Lavoisier is credited with the discovery of the stable properties and artificial formation of magnetic oxide, and Mr. Robert Mallet, M. I. C. E., also discovered it during one of his experiments.

Berthier noticed it also when iron was exposed to highly heated air, and Thirault is said to have produced Fe_3O_4 in 1860 in a peculiar manner, his process being now employed by gunsmiths to oxidize their work.

The use of oil and hydro-carbon vapors injected into the chamber or muffle where the articles of iron were under the process of oxidation by decomposed steam has received considerable attention from experimenters, and in general with favorable results, particularly when the articles in the muffle were rusty when put in, and it was necessary to change the Fe_2O_3 into Fe_3O_4, the condition being that the steam must be admitted continuously during the admission of the oil, and that the air must be excluded from the muffle, the oxygen from this source not being as effective to produce good results in the formation of the magnetic oxide as that derived from the steam; the presence of the hydrogen, no doubt, contributes to the success of the operation. Badly corroded cast-iron water-pipes have been thus treated with success, the process removing all the rust cones, blisters, etc., and rendering the pipe, so far as experiment has thus far determined, as good as, if not better than, new pipe with the ordinary coal-tar composition coating.

THE GESNER RUST-PROOF PROCESS.

This process for the preservation of iron and steel is of comparatively recent introduction, and, like the Bower-Barff process, forms a coating from the metal itself that it is depended upon to protect.*

While the Bower-Barff process forms a magnetic oxide coating, the Gesner process claims to change the surface of the metal into a compound of hydrogen, iron and carbon, which is designated as a double carbide of hydrogen and iron; analysis of treated articles showing the presence of hydrogen in quantities from $\frac{2}{10}$ of 1% and upwards. This coating does not alter the dimensions of the articles treated, screw threads and nuts running together as freely after treatment as before; while the Bower-Barff

* United States patents to George W. Gesner, No. 376,874, January 24, 1888. Apparatus for oxidizing the surfaces of metals; described in the *Official Gazette*, vol. 42, p. 343. No. 379,866, March 20, 1888. Treatment of metals; described in the *Official Gazette*, vol. 42, p. 1,268.

See *Iron Age*, April 3, 1890, for an illustrated article of the process and apparatus; also the *Engineering Journal* (Forney's), April, 1891, for description of the process.

process, when carried to the point of forming a heavy coating of magnetic oxide, materially increases the size of the parts, and has to be allowed for in fitting up of finished and jointed parts, such as hinges and loose joints.

A claim made for articles treated by the Gesner process that they are unaffected in strength appears to have some foundation.

The record of a number of tests made at Philadelphia on an Olsen testing machine show that the wrought iron samples were comparatively unaffected in strength by the treatment, while the steel articles showed a loss in strength of about 5%, and a gain of 5% in elongation, evidently from the annealing action of the muffle. Cast-iron pipe treated, shows that the pores of the iron are filled, and the strength materially increased. Cases reported show an increase in bursting pressures of over 100%.

The color of the treated articles is a fine deep blue-black. Tempered articles, such as springs, bayonets, edge tools, etc., cannot be treated, but in some cases articles can be hardened and tempered after treatment. The process does not necessitate the use of skilled labor; after being established, ordinary working men are all that is requisite. The process is adaptable wherever tinning or galvanizing are employed to withstand atmospheric effects, and is materially cheaper than those methods. Among the adaptations for this process can be named: office railings, grilles, gates; wrought, cast and malleable iron pipes and fittings; steel and iron shingles, corrugated sheets for roofing, polished wrought-iron pipe for bedsteads, steam radiators, builders' and art hardware, architectural iron work of all kinds, gun fittings, etc. These articles can be treated in small lots for prices from four and a half to seven cents per pound, and large pieces, wrought iron pipe in 20 feet lengths in quantities, will not cost over one cent per pound, including manufacturer's profit. One of the principal articles thus far treated, is a clock tower on a storage warehouse on Schermerhorn Street, Brooklyn, the clock dials of which are of wrought-iron scroll work, 16 feet in diameter. The surface of the treated articles appears to facilitate the spreading and adhesion of paint, if any is subsequently required on the structure.

The articles must be cleaned from scale to secure a good result, though the removal of oil and grease due to machining processes is not necessary.

The apparatus and process of treatment is viz.:

Two ordinary clay gas retorts of any required length from 7 to 20 feet or more, are set side by side with a furnace and grate below and between them, same as a gas retort setting. Suitable doors are fitted to one or both ends of the retorts as used for gas retort work. These retorts are heated to 1,000° or 1,200° Fahr., as may be determined by the character of the articles to be treated. The articles are placed in the retorts, and exposed to the heat for

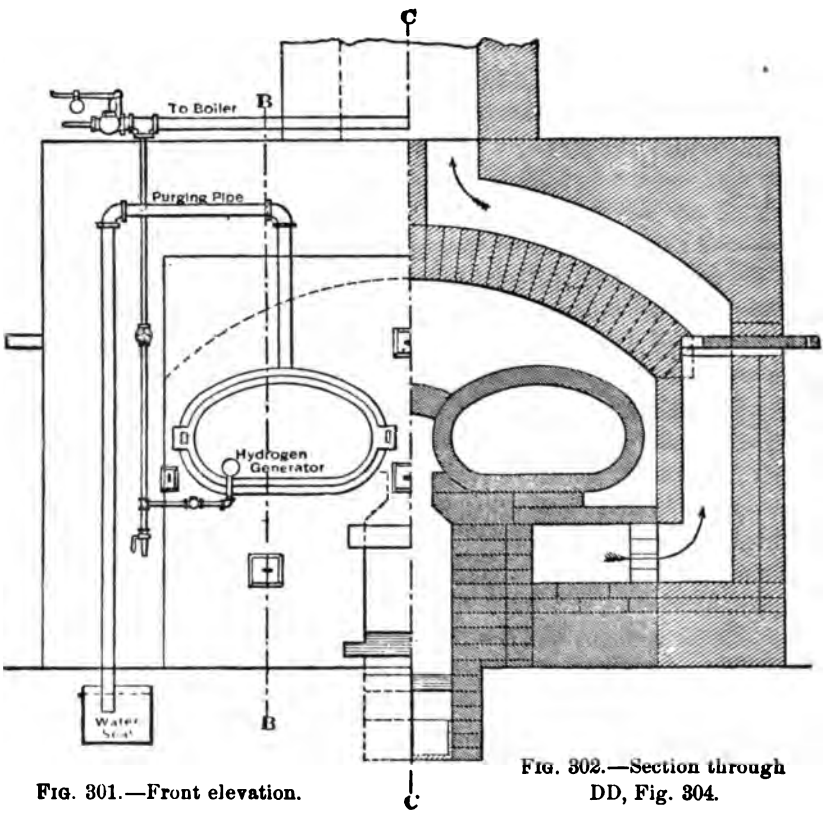

FIG. 301.—Front elevation.

FIG. 302.—Section through DD, Fig. 304.

20 minutes or until they attain approximately the heat of the retort or muffle, when steam at low pressure is admitted to the inside of the retort through an iron pipe lying on the bottom of the retort, and called a hydrogen generator, which decomposes the steam, hydrogen being set free, which fills the retorts, the surplus pressure passing out by a purge pipe sealed in about 1½ inches of water to prevent ingress of air to the retorts.

This steam treatment is kept up for 35 minutes, when a small quantity of naphtha—a pint or more—is injected by gravity into each retort, and allowed to flow for 10 minutes. The hydro-carbon is then cut off, and the steam, which has been allowed to flow during the whole operation, is continued some 15 minutes longer.

The whole time employed is therefore an hour and 20 minutes for articles of ordinary size and weight, such as builders' hardware, etc.

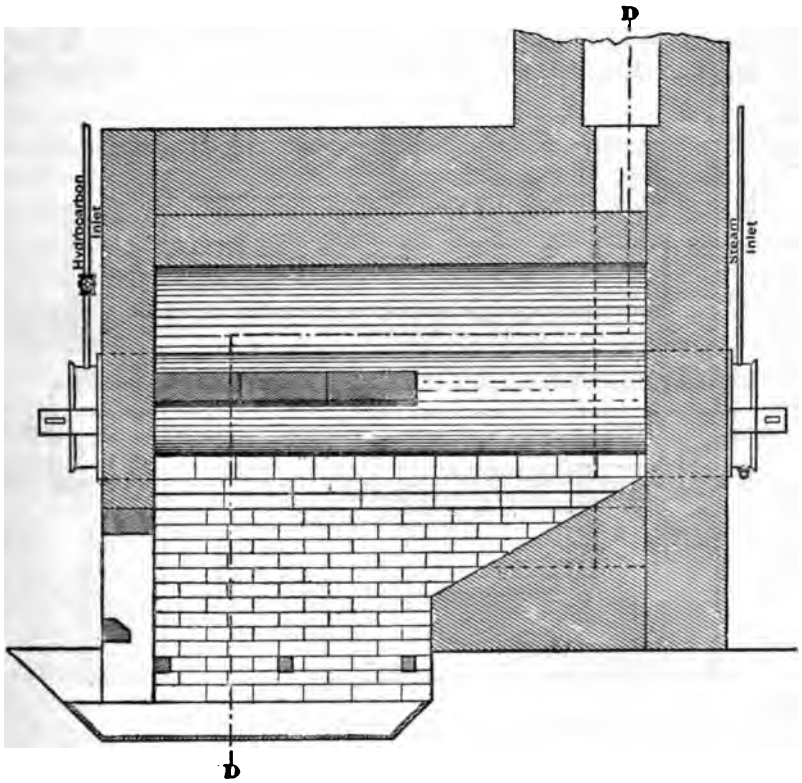

FIG. 303.—Section through BB, Fig. 301.

The articles remain in the muffle sealed from the air until the heat has fallen to about 800° Fahr., when they can be removed without danger of scaling.

Ornamental objects, art hardware, etc., while hot are given a bath in whale or paraffine oil, to render them even in tone, prevent finger marks, etc. The process appears to be almost identical with the Bower-Barff, with the exception of the use

of the naphtha, which gives a foundation to the claim for the formation of a carbide that hastens the process; the time of treatment being materially less in the Gesner process than by the Bower-Baff to get the same thickness of coating. A number of plants for the working of this process are in operation in various parts of the United States. The principal one, the South Brooklyn Rust-Proof Iron and Steel Works, 176 Sullivan Street, near Atlantic Basin, South Brooklyn, has a capacity of two tons, daily output, of small articles like builders' hardware, and six tons of pipe up to 7½ feet in length, and can treat articles up to 20 feet in length. Members of the American Society of Mechanical

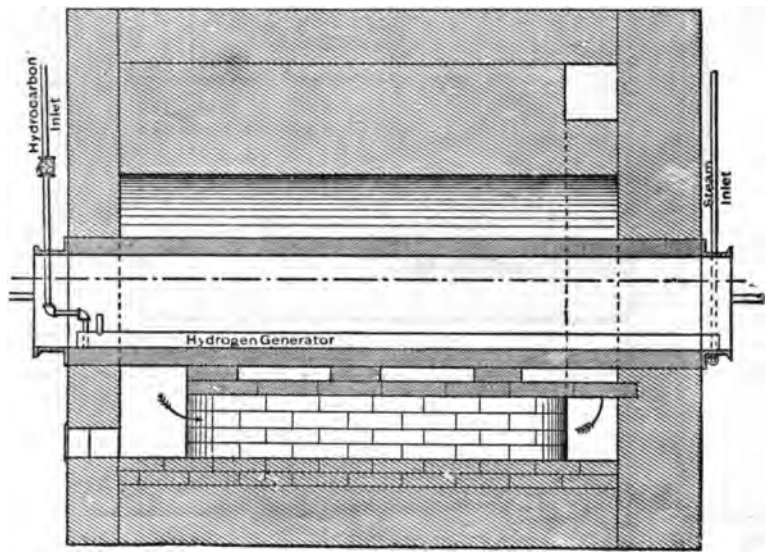

FIG. 304.—Section through CC, Fig. 301.

Engineers are welcome at any time to inspect the works and product, Mr. Geo. W. Gesner, proprietor, being in charge of the plant. A number of samples of rough surface and polished work are submitted for inspection.

The *London Engineer*, dated February 9, 1894, has a most excellent article upon the "Rusting of Iron and Steel;" though it recommends no remedy or process, it is so valuable a contribution to the literature upon the subject that I quote it entire.

"The phenomena of chemical combination appear to be exceed-

ingly complex. Not so many years ago we were taught that a mixture of oxygen and hydrogen would combine to form water when an electric spark was passed through them. The matter appeared simple, was easily expressed in chemical formulæ, and illustrated by experiment. Now we have learned that it is impossible to make such a mixture explode when it consists of perfectly pure and dry gases. When, however, the slightest trace of moisture is present the combination takes place at once, thus illustrating the importance of those 'next to nothings' which were so ably discussed by Sir Frederick Bramwell in his address to the British Association. The oxidation of iron, though a more familiar phenomenon, is at least as complex as that of hydrogen. In spite of the proverb, this familiarity has been far from breeding contempt, as its commercial importance has attracted considerable attention to the subject, and though there is much to be learned, some few facts appear to be now established. In the first place, neither bright iron nor steel will rust in pure water or in pure air. The presence of carbonic acid or some similar agent seems necessary, although the final product may be destitute of carbon. Even when oxygen, moisture, and carbonic acid are all present, rusting will not take place unless the moisture condenses on the surface of the metal.

"When rusting does take place under ordinary circumstances, the first stage appears to be the formation of ferrous carbonate.

"The carbonate is then dissolved in carbonic acid water to form ferrous bicarbonate, which latter is then decomposed in presence of air and moisture to form hydrated ferric oxide, magnetic oxide being formed as an intermediate product.

"This fact, as to the formation of the magnetic oxide, is curious, as the Bower-Barff process of protecting iron and steel consists in coating the metal with a firmly adherent layer of this very oxide.

"Every one knows that when a bar of iron has commenced to rust, the corrosion proceeds apace. A polished bar will resist oxidation for a comparatively long time, even under somewhat unfavorable conditions, but once the rust has commenced to form it does not take long for it to cover the whole bar.

"One reason for this may be the fact that the rust is electropositive to the iron, but it is also partly attributed to the final product, the hydrated ferric oxide, being only formed at the end of several intermediate stages of the oxidation, and to its hygroscopic properties, which favor the absorption of moisture from the

air. In certain situations other acids besides carbonic may take part in the corrosion of iron. The metal work in bridges over railways is particularly exposed to fumes, and some engineers consider that in such cases no plates less than ⅜-inch in thickness should be made use of, even in the case of the flooring.

"The whole question of the rusting of iron and steel work has been discussed in considerable detail by Mr. Thomas Turner, Assoc. R. S. M., F. I. C., in a paper recently read before the South Staffordshire Institute of Iron and Steel Works Managers.

"It is now pretty generally acknowledged that, so far as ordinary exposure to the weather is concerned, iron is less liable to rust than steel. Unfortunately, however, this capacity for resisting rust seems to be greater in the common irons than in the best qualities, and has been attributed to the phosphorus contained in the former, which seems to have a protective action.

"When iron and steel are used in conjunction there is no certainty which will be the more liable to rust. The potential difference of contact between the two is very small, and though in general wrought iron is found to be electro-positive to steel, there seem reasons for believing that this may not be so in all conditions.

"Mr. W. Denny has instanced a case where the steel shell plates of a vessel remained clean, while the iron stem plate and rudder forgings were much corroded.

"The ballast tanks of ships are particularly exposed to rust. Bilge water is an exceptionally powerful corroding agent, and several engineers have suggested the use of iron plating in ships, in those parts exposed to bilge water, even when the body of the ship is of steel. In steam boilers it is claimed that there is little difference in the behavior of the two metals, and certainly steel boilers, when properly looked after, have proved to have a long life.

"Plates thoroughly cleaned from scales are less liable to corrosion than when used just as they come from the rolls; and the Admiralty have accordingly adopted the practice of pickling the plates before being used. In case of pitting, Mr. John found a particle of black oxide at the bottom of each pit.

"Experiment shows that this black oxide is strongly electro-positive to the plates.

"Cast-iron seems in general to last better in sea-water than either wrought-iron or steel. Trautwine, however, relates that

the cast-iron cannons of the Royal George and the Royal Edgar, after an immersion of 62 and 133 years respectively, had become quite soft, and were in some cases like plumbago. A very similar experience was noted with the cast-iron sluice-gates of the Caledonian Canal. Much apparently depends upon the quality of the iron.

"Trautwine recommends white, close-grained cast-iron, while Mr. Turner quotes from a British Association Report recommending gray iron.

"The alloys of iron with nickel, cobalt, and chromium appear less liable to rust than ordinary iron, while the presence of manganese appears to render the iron more sensitive to attack by corrosion."

Commenting upon the above in its reference to steam boilers, I think the presence of the scale found in most boilers acts as a partial preservative, as in general the shells of all types of boilers show the effects of corrosion more upon their external than internal surfaces. Boilers using perfectly pure or distilled water are more liable to pitting than a general corrosion over the whole surface of the shell and tubes. Why this is so is a mooted question. Pure hot water is a strong solvent of nearly all bodies; steel and iron do not escape its action, as the collapse of the Boston, Mass., hot-water circulating system forcibly shows, and is fresh in the minds of all engineers. The Heine boilers that furnished the hot water, under a pressure of 450 pounds, for the system in question, were comparatively unaffected, and other boilers and pipes under similar conditions have worked for years without showing the tendency to melt away as did the street pipes in this case.

The course of the Admiralty in causing all boiler-plates to be pickled and cleaned from the scale of manufacture, is in the right direction, and should be extended to all the plates used in the construction of ships. In fact, for all iron and steel requiring protection from corrosion other than by the magnetic oxide processes, and that depend upon paint for their protection, it is an indispensable condition that the scale must be removed in order to secure the best result. When this is done and strictly pure red lead is mixed with pure raw linseed oil that has escaped the manipulations of the whale or menhaden oil merchant, a paint is produced that will unite so closely to the iron or steel surface as to be only secondary in preservative qualities to magnetic oxide in resisting

atmospheric effects. The writer cites from his experience a number of wrought iron tie bars that in an outdoor exposure of over fifty years were as unaffected from rust as though they had been up only a week. Examination showed that the bars had been scaled before painting, and had received two coats of pure red lead and oil paint at the date of their erection, and from their location in the structure could not have been repainted without removal from their position.

How rapidly rust will form beneath the scale of manufactured iron, one need but to look at the trusses and iron work of the elevated railways of New York to recognize the importance of this preliminary step in the protection of the surface. In this case successive coats of a poor, cheap paint are spread at intervals over the whole easily accessible parts of the structure; no efforts being made either by the engineers or painters to remove the blisters caused by the corrosion in progress beneath the first coat. How long this process can continue without seriously weakening the structure, is a matter of guess-work.

A recent inspection of the elevated railway lines shows that the structures below Fifty-ninth Street are, in general, corroded more than above that point, which can be attributed to the greater quantity of sulphurous gases from the numerous fires in the lower portion of the city; also to the influence of other deleterious gases that gain ready access to it from its less height as compared with the upper portion of the line, as well as the more confined space in which these gases act, owing to the many street crossing pans, that, as it were, house in the structure.

What has been the composition of the various coats of paint that have been applied to these structures, the writer is of the opinion that no one knows; but the shades of Nile green and other delicate colors that have graced them from the first, appear to indicate that the æsthetic effect has been more sought for than the protective.

At many points on the line the structure is in that condition that the old coatings must be burned or scraped off, and the iron surface be well cleaned, in order to insure a continued life to it, as the application of a new coat would do but little to protect the iron from decay, but only serve to cover the stains due to the corrosion in progress beneath the several coats of paint now in place.

The potency of iron oxide to destroy paint is well illustrated

on these elevated railway lines near the several stations, where the iron ground off from the brake-shoes, from the frequent stoppages of the trains, has settled upon the external surfaces of the structure, oxidized, and is rapidly destroying the paint in minute spots, that not only permits the moisture from the air access to the metallic surfaces, but charges that moisture with a solution of iron oxide and sulphurous water, both active agents of corrosion.

The advocates of iron oxide paints as a protective agent against corrosion, on the theory that the pigment is held to the surface of the coating by the liquid or oil mixed with it, the latter forming an elastic medium or bond between the metallic surface and the pigment, *can possibly* find here a good example of *the folly* of their claim. This extra external coating being a watery solution, instead of an oily one, carries and deposits the iron oxide all the same, and is hastening the decay of all material with which it comes in contact.

A suggestive lesson upon this point is found in the collapse of an iron and slate roof, covering the retort room of one of the New York Gas Works, that suddenly gave way without warning, and played the part of an extinguisher to all beneath. The tie rods of the trusses proportioned to withstand five times the weight of roof, with the usual allowances for wind and snow pressures, had wasted away beneath successive coats of paint and coal tar applied, until rupture came as a matter of course even under conditions of a minimum strain.

The material in the old Hammersmith bridge across the Thames at London was bought by Sir William Arrol to use for false work at the Forth bridge. The iron after sixty-two years of service was as good as new. Many of the parts were inaccessible, and had not been repainted since their erection. Pure white lead paint was at the bottom of the secret.

The red houses and barns that are so familiar a feature in New England scenery, have been standing from fifty to over one hundred years exposed to an atmosphere and climate charged with all the conditions that cause rapid decay. These buildings, when erected, received, in general, two coats of red oxide of lead and pure boiled linseed oil paint, that has so thoroughly protected them, that where the paint has in many cases been completely cut away by the sand blast of the Cape Cod storms, the wood sheeting is in good condition and the nail heads are as perfect as when

first driven. But the paint was honest; no red or brown oxide of iron with a liberal addition of barytes, whiting, pipe clay, ochre, and other "*almost as good as lead*" *compounds*, mixed with sour sig and oil so fishy that one could almost detect the fish scales on the painter's brush, entered into this work.

The Puritans knew nothing about railways, steam-engines, telegraphy, telephones, the wiles of electricity, or of the Hawaiian question, and even escaped acquaintance with the A. S. M. E.; but they could paint their sheep sheds so thoroughly that the generations succeeding them have had but little to do in the form of renewals and repairs.

The *Engineering Journal* (Forney's), Vol. 65, No. 4, page 163, April, 1891, has a long and instructive article on "The Preservation of Iron and Steel Structural Work," by Woodruff Jones, A.M., that embodies so much useful information upon the subject of paint and its application to metallic surfaces, that I quote it almost entire in order to bring the best productive methods of the day down to a point for comparison:

"The requisites of a good paint for the preservation of iron and steel are, that it shall adhere firmly to the surface and not chip or peel off; it must not corrode the iron, else the remedy may only aggravate the disease. It must form a surface hard enough to resist frictional influences, yet elastic enough to conform to the expansion and contraction of the metal by heat and cold. It must be impervious to, and unaffected by, moisture, atmospheric, and other influences to which the structure may be exposed. The paints that have been used for this purpose are principally asphalt and coal tar paints, consisting of mineral and artificial asphalt or coal tar, either applied alone or combined with each other, and more or less with metallic bases, and iron oxide paints and lead oxide paints, especially red lead, in all of which the pigment is held to the surface of the metal by combination with linseed oil. The choice of paints must lie, so far as our present practical experience goes, between these three classes, zinc oxide being found to be entirely unsuitable on account of its propensity to peel off.

"What, then, is found to be the experience in actual use with these? Asphalt and coal tar paints run when exposed to the sun and other sources of heat, which is a serious matter with vertical surfaces, and after a time become extremely brittle and scale off entirely, leaving the under surface exposed unless the paint is con-

stantly renewed. In the mean time the exposed iron and steel are being corroded by rust.

"Iron oxide paints, including 'metallic brown,' are paints made from iron ore, or by some chemical process with an iron base. These are invariably iron in a greater or less degree of oxidation, or, in other words, rusted iron. Now, it is well known that one of the most active promoters of rust or decay in iron is the rust itself. Under the combined influences of the moisture and carbonic acid of the atmosphere, iron oxide, or iron rust, becomes a carrier of oxygen from the air to the metal, rust begetting rust; it is therefore evident that this material alone has no preserving effect on iron; in fact, it promotes its decay.

"How is it when combined with linseed oil in the form of paint? In the economy of nature, iron oxide is a great disinfectant. When in contact with organic matter and moisture, even at a low temperature, under favorable conditions, it readily gives up oxygen, destroying, more or less, the organic matter, and being reduced itself to a lower oxide.

"When thus reduced, with equal readiness it absorbs oxygen from the atmosphere and again passes it on, thereby promoting and eventually insuring the destruction or transformation of the organic matter with which it may be in contact either in the soil or elsewhere. The same process appears to take place when combined with linseed oil in the form of paint and exposed to atmospheric influences, the oil being the organic matter.

"If linseed oil in drying formed an air and water proof film, it might be urged that the oxide of iron would be entirely protected from the direct influences of oxygen of the air and moisture; such, however, is not the case. The most eminent authorities have recently shown that the dried film of linseed oil, unless united with a pigment that *combines chemically and forms a water-proof coating with it*, actually absorbs water very much as a sponge. Where water will go, air will also go, and we thus have in direct contact with the iron oxide of the paint, *which does not* combine chemically with oil, those elements—air, moisture, and organic matter—which cause the iron to become a carrier of oxygen and a destroyer of what it is in contact with.

"It is well known that iron paint darkens with age; this is caused largely by the iron oxide losing oxygen, which is partly transferred to the oil, burning it up, and destroying its tenacity, as may be seen by examining iron structures painted for some

time with iron paint or metallic brown, the paint being found extremely brittle and in feathery scales.

"This is not all the damage done. The iron oxide in the paint becomes a carrier of oxygen to the metal it was designed to protect, and the process of corrosion is commenced and carried on under the paint, which eventually peels off, the surface of the metal being found more or less oxidized and corroded.

"Asphalt and iron oxide being thus shown to be entirely incapable of preserving iron, it remains for us to consider the effect of *red lead.* This pigment has the property of forming with linseed oil a *hard elastic coating* clinging with great tenacity to the metal.

"It has no oxidizing effect on iron other than to promote the formation of magnetic or black oxide that prevents corrosion, and does not act as a carrier of oxygen from the atmosphere after the paint has set, neither does it render the oil brittle nor promote rust. When red lead fails, it is principally by gradual wear or friction from the outside. It does not scale or blister, which both asphalt and iron oxide paints will do, thereby requiring a thorough scraping and removal of old material before a new coat can be applied. Any red lead pigment adhering to the metal forms a permanent base for subsequent paintings, and is utilized in further preserving the metal.

"The Government specifications for iron work in the new Library Building of Congress provide, that 'all the work not Bower-Barffed must be given one coat of pure red lead paint—*not metallic paint* of any kind, but *pure red lead*—before leaving the shop and *before* becoming rusted.

"The experiments of the Navy Department on the preservation and fouling of plates covered with different pigments are interesting. A plate of iron covered with asphalt paint was immersed in sea water for eight months and six days. At the end of that period it was found to be covered with scum and mud and very badly pitted. A plate coated with iron paint was found to be covered with shell and coral, but little paint remained, and the plate was badly pitted and rusted. A plate with two coats of red lead was found to have a few barnacles attached, but to be in fair condition, with *no rust whatever* on the iron after the paint was removed. The red lead not only protected the iron better than the other pigments referred to, but the plates were in better condition as regards barnacles and fouling. The superiority of

red lead being thus established, it is adopted for use on the hulls of all Government vessels.

"If red lead is thus proven to be the best pigment for preserving iron and steel structures, what is the best method of applying it? The value of red lead depends upon its forming a hard, elastic coating having a great tenacity for the metal. This is owing to its forming certain combinations with the oil and actually setting, very much the same as plaster of Paris or cement sets when mixed with water. To successfully work the latter substances it is necessary to put them in shape as quickly as possible after mixing with water before setting takes place. If the chemical action of setting has partly taken place the material may be moulded, but good results will not be obtained. Red lead, like these substances, must be applied to the work before it sets with the oil. It is on this point that failures in the use of the pigment have generally occurred, because if it be applied after the combining or setting process has taken place the hard, elastic, clinging coating will not be formed *on* the metallic surface.

"The following is the practice of a large ship-building establishment in applying red lead to the hulls of vessels. The plates are first pickled in a dilute solution of muriatic acid, then passed through rapidly revolving wire brushes which remove all scale and dirt, leaving the plates with a bright surface; then are thoroughly washed with pure water and rubbed entirely dry, and immediately coated with red lead and raw linseed oil. The red lead is first thoroughly mixed with just enough linseed oil to form a very thick, tough paste, which will keep for several days without hardening. This paste, as wanted for use, is thinned down to the proper consistency for spreading with pure linseed oil, and applied at once; care being taken to leave the paint-pots empty at night. A gallon of paint thus prepared contains 5 lbs. of oil and 18 lbs. of red lead, and will cover on first coat about 500 square feet, the second coat about 600 square feet of surface. In this way the red lead and oil get their initial set *on the surface of the metal*, and the closer the pigment is brought to the metal the more durable will it be found.

"Iron primed with iron-oxide or metallic brown paints before applying the red lead is a mistake, as these paints readily scale and, of course, carries the red lead with it. Iron primed with oil before applying the red lead, prevents the adhering paint from coming in contact directly with the metal, and should be avoided,

because it is out of the question for it to hold on to the surface by any process of absorption into the pores of the metal.

"The rusting of iron *before* the application of paint, which is sometimes recommended, should by all means be avoided, as it not only prevents the contact of the paint with the metal, but induces a chemical action which may go on with its corroding work under the applied paint, as referred to before.

"The application of dry lamp-black, an ounce to the pound of red lead, mixed together dry and before the oil is added, changes the color to a deep chocolate, and prevents the red lead from taking its initial set with the linseed oil as quickly as when mixed with oil alone. This compound will remain mixed in a paste form for thirty days without hardening. If rapid drying is desired, Japan dryer can be mixed with the oil used in thinning the paste before application with the brush.

"The first cost of painting structures with iron oxide is less than with red lead; however, as the labor in painting structural work costs twice as much as the material, true economy must, therefore, be sought in the durability of the paint as well as the preservation of the structure from rust.

"Structures painted with iron paints have to be repainted the third or fourth year, and if the old material be entirely removed before applying the new coat in order to secure a good application, it entails an increased cost; whereas, with red lead applied at the end of the sixth year, no such expense is necessary, but as before stated, the portions of the pigment remaining on the metal, continues to protect the surface, and is the best base for the new coat, and contributes materially toward it."

The asphalt here referred to is, doubtless, the product of coal-tar distillation, called in the trade "asphalt"; and while having a certain resemblance to natural or Trinidad asphalt in some of its physical qualities, is chemically very unlike it. The name asphalt being applied to both the natural and the coal-tar or artificial product, naturally leads to some confusion on the subject. They are, in fact, so widely apart in all of their essential qualities that they cannot be appropriately coupled together as relating to the same substance.

In the distillation of coal-tar, until the final residuum of coke is reached in the still, there are no constituent oils derived from the process that do not gradually volatilize by the heat of the sun or approximating temperatures; and all coal-tar or

hydrocarbon products suitable for use in, or as, paints also become fluid when exposed to heat; in fact, but few of them are applied in any other condition than while hot. They are all liable to run on vertical or slightly inclined surfaces, until by evaporation they are so advanced on the road to brittleness that they solidify, and by a little further progress in the same direction they become brittle and scale off on the least mechanical disturbance.

On the other hand, true asphalt paint *does not* become fluid or run on vertical surfaces when exposed to the heat of the sun or to that of boiling water. Its constituent oils are absolutely non-volatile at the highest temperature of the sun's heat, and do not change by oxidation under any ordinary atmospheric conditions. Its toughness, and adhesiveness to all bodies, wooden, fibrous, as well as metallic, are remarkably persistent and durable. Its covering quality is also excellent, and for the exclusion of moisture and prevention of rust it has no superior, if any equal. When a new coating of paint is required, it is best to apply it over the old one with but little or no scraping. The mixture of foreign substances, such as clay, talc, mica, barytes, plaster, oxides of iron, etc., to give body or some other "*don't know what*" quality, is strongly condemned. Pure Trinidad asphalt needs no bush. Cuban and other brands of asphalt have so many earthy impurities as to preclude their use for paint or preservative compounds to be applied to metallic structures. Their use should be relegated to paving and kindred purposes, or to metallic surfaces of minor importance or ground work.

How effectually a light coating of the proper material will protect a surface from decay is forcibly shown, and examples can be seen in almost every city and village in the civilized world. The swinging sign-boards of the roadside inn—in many cases exposed for nearly a century to bleaching in the sun and the buffets of storms—still show clearly the black letters of the names and symbols that were originally placed upon them. These wooden signs, on examination, prove to have been given a number of coats of linseed oil and white-lead paint, over which the lamp-black letters and symbols were placed in a *single* coat, which has not only protected itself, but all the paint coatings over which it was spread. In many cases the paint in the interstices between the lettering, and even the wood of the sign, is worn away until the letters stand out in bold relief.

These remarkable results are to be attributed to the lampblack pigment. Lampblack as a carbon is practically unchangeable and indestructible under ordinary atmospheric conditions, and being of itself of an oily and elastic nature, its combination with oil forms an elastic, close-clinging coating—one of the best preservative paints known in the arts. Its want of color is alone against its general adoption; but its mixture with other pigments to tone and modify their character is admissible and beneficial in almost every instance. It is neutral in action, non-corrosive, and cannot be excited into an electro-positive state, as many pigments are, by the addition of oxidizing materials and exposure to the elements.

Cast iron generally seems to withstand corrosion from exposure to the atmosphere, sea water, acidulated water, sulphuric and chemical fumes, better than either wrought iron or steel. Trautwine's citation of the cannons immersed in the sea for 62 and 133 years and being as soft as plumbago when taken out, can be supplemented with the fact, that they were nearly of the original size, while the anchors and chain cables removed at the same time were corroded away to but skeletons and had hardly more strength per inch of metal than the cannons. It may be noted that when the cannons and anchors were first taken out of the sea, they absorbed oxygen from the atmosphere at so rapid a rate as to become so hot that they could not be handled, a condition noticeable for a number of days. As between a gray cast iron recommended by Mr. Turner, Assoc. R. S. M., F. I. C., for iron constructions immersed in the sea; and Mr. Trautwine, who recommends white close-grained cast iron, I should prefer to follow Mr. Trautwine, but it is a question of guessing which to select. Parts that need machining will require to be softer than those that can be put into place as they come from the foundry. Accurate data is wanting on all these points. Experience is a dear school, and a slow one in its deductions. It has been noted that cast iron car wheels immersed in both fresh and sea water for many years, show less corrosion on the chill part of the wheel than on the body. The iron is necessarily the same in both points, but the molecular arrangement of the iron, due to the chill, not only imparts a higher resisting power to wear, but also to corrosion. This example shows that both Trautwine and Turner are right, though diametrically opposed to each other primarily, and the governing effect of a condition rather than a theory.

Professor V. B. Lewis of the Royal Naval College, Greenwich, England, at a recent meeting of the Institute of Naval Architects, London, states:

"The rusting of iron and steel is a definite chemical process due to the conjoint action of air, moisture, and carbon dioxide upon the metal. The increased rate of chemical corrosive action due to a local increase of temperature is noticeable, and may be due to galvanic action set up between portions of the same metal at different temperatures.

"It is an undoubted fact that the double bottom of ship plates near the boilers corrode more rapidly than similar plates in other parts of the vessel, and the increase in temperature near the boiler is the only factor.

"It is also noteworthy that the plates at the bottom of the cellular spaces which are kept cool by contact with the sea-water do not corrode; and cases are noted in which parts of a plate, which get locally warmer than other parts—although the difference can only be a few degrees—corrode much more rapidly than the cooler portions.

"Experiments show that the rapid corrosion found in the double bottoms near the boilers or other sources of heat, is due to galvanic action, and not to the increased chemical activity due simply to the increase of temperature. As the ashes are drawn and quenched with sea-water near these exposed plates, no doubt some of the corrosion can be traced to the gases thus formed; the sulphur in the ashes also contributing its effect."

Mr. William Thomson, F.R.S., F.I.C., read a paper, "On the Influence of some Chemical Agents in producing Injury to Iron and Steel," before the Manchester Association of Engineers, November 25, 1893, in which he refers to the interesting and exhaustive experiments made by Mr. Thomas Andrews, F.R.S., Ed. Assoc. M. Inst. C. E., on the galvanic action which takes place between iron and steel, and between iron of different kinds and steel of different kinds.

One paper, entitled "The Galvanic Action between Wrought-Iron, Cast Metals, and Various Steels during Long Exposure in Sea-Water," read before the Institute of Civil Engineers, Vol. 77, Session 1883-84, Part III.; the second paper, "Corrosion of Metals during Long Exposure in Sea-Water," read before the Institute of Civil Engineers, Vol. 82, Session 1884-85, Part

IV.; and the third paper, "The Relative Electro-Chemical Positions of Wrought-Iron, Steel, Cast Metal, etc., in Sea-Water and Other Solutions," read before the Royal Society of Edinburgh; also, a paper by Mr. David Phillips, read before the Institute of Marine Engineers, May 19, 1890, which shows that steel corrodes more easily than iron.

These papers are well worthy of perusal and a permanent place in any engineer's library. I regret that their length forbids their introduction into this article, or even an attempt to abridge them.

In the above named articles Mr. Andrews shows that while some varieties of iron and steel remain constantly electro-positive or electro-negative to each other, others change, taking opposite positions toward each other; while others again change positions constantly during long periods.

"It can be easily understood that while there is no material voltaic action between two pieces of steel or two pieces of iron, or of pieces of steel and iron, there may be conditions on the surface of one plate or rivet which may act strongly as an electro-negative element, and produce rusting on the metal in contact with it. A piece of iron immersed in *weak* nitric acid begins to dissolve at once. A similar piece placed in *strong* nitric acid, touching it for a few minutes with a piece of platinum wire, and then putting it into the weak nitric acid, will not dissolve, it having been rendered passive; and similarly, there is reason why one piece of iron may act electro-negatively toward another piece of the same metal, on account of some slight alteration of its physical properties, by hammering, such as closing the riveted seams of plates, calking seams, setting tubes, etc.; or it may have attached to it some peculiar oxide of iron (for there are a large number of iron oxides) which may act electro-negatively toward any metal with which it is in contact, and may thus induce oxidation in such metal."

"Chemically pure iron will not rust in chemically pure water. For rusting to take place, it is necessary to have another substance which is electro-negative to the iron to be in contact with it, so that the current of electricity liberated by the oxidation of the iron passes away to the metal, or other material which acts as the electro-negative element. In this way the iron acts as one of the elements of a voltaic cell."

A plate of iron and one of copper joined together or in com-

munication mechanically, and immersed in sea-water or acidulated solutions of water, the iron becomes electro-positive to the copper and the iron is rapidly corroded. If the iron is connected to another metal which is electro-positive to the iron—zinc, for instance—then the iron becomes electro-negative and remains comparatively unaffected, the zinc being the metal corroded. The metal that is wasted away will always be found in communication with the positive pole or current of electricity. The fluid in which the iron and copper are immersed will be found colored red with the iron rust, while that in which the iron and zinc are placed will contain a white flocculent precipitate of zinc that has almost prevented the iron from corrosion.

This system of preserving iron from corrosion in steam-boilers has been patented in England and America, and is in extended use under a number of different applications. The zinc aids materially to prevent incrustation, because, as the zinc oxidizes, the hydrogen is evolved on the surface of the iron in small quantities, and this aids in preventing the sulphate and carbonates of lime and magnesia thrown down from the water from adhering to the plates as a hard scale. On a piece of iron that has become corroded by oxidation, it will be found that the corrosion has taken place in small holes or pits, technically known as "pitting"; these are produced by some impurity in the iron which ultimately forms the centre of the pit; this may be a piece of carbon, a speck of manganese, or other substance which is electro-negative to the iron, which latter being electro-positive becomes oxidized. Steels that contain large amounts of manganese are peculiarly subject to oxidation.

When rust begins to form on iron, it usually attacks it at certain minute points and extends, like spots of mould, the oxide of iron itself acting as the electro-negative element to the iron on which it rests, so that when a piece of iron has become rusty it is very difficult, after cleaning, to prevent it from again becoming rusty unless every particle of rust be most carefully removed from it, each particle of rust forming an electro-negative element around and under which the electro-positive iron begins to oxidize and produces a small hole or pit. When the magnetic oxide coating of an article treated by the Bower-Barff or other kindred processes is broken, the coating of black oxide then acts as the electro-negative element, and the iron is rapidly corroded into red oxide of iron, or rust, as much energy in the

form of electricity and heat being developed in the combination of the oxygen and iron to create the rust as was expended in the process of smelting the iron from its original oxide ore into metallic iron.

"An important question presents itself to boiler-makers, whether it is safe to rivet steel plates to iron plates in steam-boilers, or even in other constructions, particularly where exposed to high temperatures or frequent changes of moderate temperatures; or to use iron rivets for steel plates, or steel rivets for iron plates."

"Cases are shown where, furnace plates of steel riveted together with iron rivets, the plates in the vicinity of the rivets are badly rusted or pitted while the rivets remain intact. To determine how iron stands to steel and how different samples of steel stand to each other, Mr. Thomson made an extended series of experiments, using a Thomson's tangent galvanometer to measure the electrical currents generated in the corrosion of iron and steel, both singly and in connection with each other, and when immersed in different fluids; viz.: sulphuric acid (one part to nine of water), caustic potash solution (specific gravity, 1.311), and chloride of ammonium solution (specific gravity, 1.033), the latter representing electrically the ordinary concentrated water found in steam-boilers."

The details of the experiments are very interesting, but I will only take space for the results obtained.

"An iron rivet and a piece of the above-mentioned corroded steel furnace plate, when placed in contact and immersed in the weak sulphuric acid bath, at first the steel was electro-negative to the iron, but in a few moments it changed, and afterward the iron was electro-negative to the steel. When placed in the chloride of ammonium solution, at first the iron was strongly electro-positive to the steel, and afterward became *weakly* electro-negative. When placed in the caustic potash solution, the steel was strongly electro-positive, but the current gradually became weaker and weaker until it practically ceased. A new steel rivet in an iron plate, a steel rivet closed by a machine and held until nearly cold, an iron rivet closed on a mild unworked steel plate, all reacted strongly among themselves. The iron when first brought into voltaic contact with the steel was strongly electro-positive to the steel, being presumably strongly acted upon by the solution, but after a few minutes almost

ceased action or became reversed; and so far as the tests demonstrated as a whole, it was to the effect that it was quite as safe to bring iron and steel in close mechanical contact with each other as two different kinds of steel or two kinds of iron."

"Dr. Frederick Grace Calvert's (F.R.S.) experiments many years ago determined that bright iron placed in an atmosphere of dry oxygen or of dry carbonic acid would not rust; when put in a damp atmosphere of oxygen it rusted slightly; in a damp atmosphere of carbonic acid a small quantity of white carbonate of iron was formed on the surface of the bright metal, but no rusting took place; when, however, bright iron was placed in a damp mixture of the two gases (oxygen and carbonic acid), it became rapidly oxidized, forming a copious excrescence of red oxide, or rust, all over it."

Experiments in the opposite direction, to prevent rusting, showed that a strong solution of carbonate of soda preserved steel needles and instruments bright and untarnished after an exposure of twenty-five years.

"Mr. Thomson's experiments with red lead, white lead, zinc white, and oxide of iron paints, and of varnishes composed of bitumen alone, and mixed with lime, coal-tar, and other substances to prevent corrosion of iron, in which plates coated with the various paints were immersed in sea-water in separate vessels. After a number of days the waters became first turbid, and on longer exposure precipitates of the red oxide of iron, or rust, were thrown down in copious amounts, the plate coated with red lead alone being unaffected. As red lead of itself is a highly oxidizing body, one would expect that instead of preserving iron it would only hasten its oxidation. The reason for this preservative action is found to be that the oxidation of some of the protoxide of iron which was first formed upon the surface of the metal was further oxidized by the oxygen in the red lead and converted into the black oxide of iron. In other words, the effect of red lead upon iron is to produce a skin of unoxidizable black or magnetic oxide on the iron itself under the paint, being the same substance as is produced by the Bower-Barff process."

Experiments show that manganese dioxide ore, or pyrolusite, being a highly oxidizing substance, forms a paint in some respects superior to red lead, as it does not harden so quickly or require to be used so soon after being mixed with the oil. Iron and steel painted with manganese paint and remaining coated

for a number of months, and then having the paint removed and the articles placed in sea-water, are found to be passive to corrosion for a long time, thus showing that the manganese has formed a coating of black oxide upon the surface of the metals, ensuring their protection from corrosion for a limited period even without the additional protection afforded by the paint itself.

Bichromate of potash, chromate of lead, and most of the chlorates, possess the peculiarity of forming the black or magnetic oxide on the surface of iron and steel to which they may be applied as a paint. The *red oxide of iron paints do not* possess this feature, but are so porous as to readily admit the passage of air and moisture, in the electrolysis of which the oxygen attacks the iron to produce rust, the hydrogen being set free to form blisters, the presence of which but hastens the process of corrosion by exosmosis of the hydrogen through the blistered paint, air and moisture taking its place and perpetuating the process of corrosion.

The use of perfectly pure materials cannot be too strongly insisted upon in the preparation of any paint applied to any important iron or steel structure. Adulteration is the order of the day, and in many cases the adulteration of the adulteration comes in to gather an extra penny from a pound of material.

There are some seventeen well-known siccative or drying oils of a vegetable origin available for the purposes of paint.

Linseed from its many qualities stands preëminently at the head of the list, and though all of the other siccative oils will dry on exposure to the air, many of them in so drying do not fail to crack, and thus defeat the object and intention of their use for preservative methods.

The number of non-drying oils of a vegetable character that are available for the adulteration of linseed oil are over thirty; the greater number of which are commercially cheaper than linseed; and as most of the siccative oils are also cheaper than linseed, the inducement of the oil dealer to give the linseed tank a dose of some one or other of the group is not always resisted. When to the above list are added the fish and animal oils, one may wonder how any linseed oil gets into the market in a condition fit to apply as a vehicle for the required paint, even when fortified for drying by the use of turpentine, Japan-dryer, benzine, etc.

There are many methods of detecting the adulteration of oils; that by the hydrometer appears to be the least satisfactory.

Hydenreich and Penot's method is based upon the fact that the several fat oils change color when brought into contact with strong sulphuric acid. When a drop of this acid is added to eight or ten drops of an oil placed on a plate of glass resting on a piece of white paper, the following colors are immediately produced:

Linseed oil becomes brown-red, changing to a black-brown. Hempseed oil becomes a distinct emerald-green tint. Poppy oil, a pale yellow with a dingy gray look. Colza oil, a greenish-blue aureola. Sesame oil, a bright red. Olive oil produces a deep yellow tint, gradually becoming green.

The above colors are produced from chemically pure oils. The colors produced by adulterated oils are too numerous to attempt to describe here. The experimenter will find the pursuit of knowledge in this branch of engineering a fascinating subject, and one pretty sure to ground him in the conviction that most oilmen are ineligible for the kingdom of heaven, and fair dealing abideth not in them.

Prof. F. C. Calvert's experiments indicate that caustic soda (specific gravity, 1.340), added to an oil previously treated with sulphuric acid (specific gravity from 1.475 to 1.635) or nitric acid (specific gravity from 1.180 to 1.33); by the reaction, it is possible to ascertain the presence of 10% of any given oil in many cases of adulteration.

Watts' *Dictionary of Chemistry*, vol. iv. p. 183, gives a table of a number of these reactionary colors with acids of different strengths, that is highly interesting and instructive. Further details on this subject will be found in *Pharm. Journal*, Trans. 13, p. 536; also, see Ure's *Dictionary of Arts*, iii. 300.

The German chemists have been active investigators in the field of oils and oleaginous compounds. Their deductions are too voluminous to attempt to incorporate herein, but they generally lead to the data given above.

The oil or spirits of turpentine or "turps" is used to a great extent in all kinds and brands of paint to aid the drying qualities. It consists of a hydrocarbon oil ($C_{10}H_{16}$) and a resin called colophony. Its action upon red lead, pyrolusite, or other of the pigments that act electro-negatively upon iron or steel to

form magnetic oxide upon the surface of the metal, is purely a mechanical one as a dryer. Its evaporation leaves behind the resin mixed with the pigment, and by its presence prevents in some measure the chemical combination of the oil and the red lead or other pigment. If it is of that kind of turpentine known as "fat," it contains so much resin of a hard, vitreous character that, when dry, it cracks into small sections and ruptures the coating of oil and paint with which it is incorporated, and materially aids the process of corrosion by admitting the air and moisture to the surface of the metal. Its presence in any form or amount in a preservative coating for iron acts in a great measure as caustic or quick-lime added to hydraulic cement. The lime will delay the setting of the cement and detract from its strength. The "turps" will hasten the drying of the oil; also detract from its resisting and protective power.

It is sometimes advisable to use a dryer to aid the drying of oil in cold or damp atmospheres; also to facilitate the spreading of the paint over very cold surfaces, etc. In these cases better protective results are obtained by the use of Japan-dryer. This dryer is made by digesting gum-shellac with linseed oil until it becomes thick and partakes of the nature of a varnish. Litharge (the scum of melted silver ore, a protoxide of lead, PbO, containing approximately 95% of red or yellowish oxide of lead) is added to quicken the drying of the resulting Japan. When it has cooked to a thick mass, termed a "pill," it is cooled and thinned with turpentine and is ready for use.

This dryer enters into a chemical combination with linseed oil and the red oxide of lead and other pigments that form magnetic oxide coatings on the surface of the metals over which they are spread; and, as will be seen from the brief of its manufacture, its principal ingredients are of the same substances that compose the protective coating for the metal, and does not retard that action called "setting of the paint" in any degree, but hastens it.

Benzine of any grade, gravity, or kind, though a perfect solvent of all the siccative and other oils of commerce, should never be tolerated in any paint compounds applied to metallic structures, and, in fact, to none even of wood. It evaporates quickly and acts solely as a mechanical dryer. It does not enter into any chemical combination with either oil or pigment, and as the redistilled and

purest samples of it still carry some appreciable quanties of paraffine in solution, the evaporation of the hydrocarbon fluid leaves the paraffine behind to act as a wax coating on the surface of the metal, and prevents that close union of the pigment with the metal so essential to a successful application of the paint. A slight change of temperature causes a blister to form on either the iron or wooden surface coated, and the process of dissolution has then commenced.

If two pieces of rough surface *scaled* iron be taken, and one is washed with benzine and allowed to dry as ordinarily practised in the workshop, or even wiped dry with ordinary cotton waste; and the other piece be washed clean with carbonate or sal-soda; and both pieces are then coated with two coats of the same pure red lead and linseed oil paint, and thoroughly dried for a number of days, and then the pieces be immersed in separate vessels filled with sea-water, it will be found at the end of three months' exposure that the sample washed with benzine has commenced to rust and pit and has thrown down a considerable amount of red oxide of iron, coloring the water very materially, while the soda-treated sample has not even discolored the water and the surface of the metal is found clean and free from rust. In the former case, the thin film of paraffine wax from the benzine washing has prevented the formation of any magnetic or black oxide of iron on the surface of the metal, but, acting as the electro-negative element and aided by the exciting sea-water, has decomposed the organic element (the oil) and has hastened corrosion not only of the metal but of the paint also.

Of the five oxides of lead, the red oxide, minium, Pb_3O_4, or red lead, is the one that principally concerns us to consider as a pigment, its action upon iron and steel being so remarkable as to place it at the head of all other pigments when applied for the purpose of preserving iron and steel from corrosion however induced. When prepared for analysis, or when the commercial article is freed from the protoxide of lead by digestion, it contains 90.63% of lead and 9.37% of oxygen, numbers agreeing exactly with the formula, Pb_3O_4. Its specific gravity ranges from 8.6 to 8.94, and as a compound it is classed analogous to the magnetic oxide of iron, a substance which red lead creates when brought into contact with iron or steel, either in a dry state or as a pigment. Red oxide of lead is found native in

many parts of the world, where it accompanies cerusite. The commercial article is generally prepared from melted metallic lead by oxidation in special furnaces, the process being a double one; the first process being to oxidize the metallic lead into *massicot* or *litharge*, which is then ground to a powder and washed. The second process consists in reëxposure of the *ground massicot* to a low red heat in presence of air for some 48 hours, or until the proper color of a dark red when hot is obtained, which changes to a bright red on cooling, the quality of the product depending in a great measure upon the latter stage of the manufacture. Commercial red lead contains all the foreign metallic oxides—such as the oxides of silver, copper, and iron, with which the *massicot* or *litharge* from which it is prepared is contaminated. These foreign oxides are, however, so small in amount as not to seriously affect the quality of the red lead as a pigment in its peculiarity of forming magnetic oxide or an electro-negative coating on iron or steel which is coated with it.

Red lead is adulterated with red oxide of iron, boles, or brick dust. These substances remain undissolved when red lead is digested in warm dilute nitric acid; boiling hydrochloric acid extracts the sesquioxide of iron from the residue. When red lead thus adulterated is ignited, there remains a mixture of yellow lead oxide and the red substances that have been added to it. The brightness of the color of red lead prevents the addition of many adulterants such as plaster, chalk, etc., the color being sensibly changed by small additions of these and like substances, as well as the weight of the lead.

Of the other substances that are remarkable for their tendency to become electro-negative to iron and steel, and to form magnetic oxide upon their surfaces when placed in contact with them, manganese dioxide, MnO_2 ore, or pyrolusite, ranks next to lead in qualities as a pigment, its oxidizing powers being greater than red lead; but the chemical combination with linseed oil and the development of the "setting power" is *not* apparently as effective as red lead. It consists of red oxide of manganese, 84 to 87 parts; oxygen, 11.45 to 14.58; sesquioxide of iron, 0.40 to 1.30; baryta, 0.67 to 1.20; silica, 0.51 to 0.80; and traces of alumina, lime, and water, etc. Specific gravity when pure, 4.819 to 4.97. Lustre, metallic; color, iron-black, dark steel-gray, sometimes bluish.

Its application as a pigment for the preservation of metallic surfaces from corrosion being comparatively recent, but little data

is available for comparision with red lead. Its color, in many cases, will lead to its application when the color of red lead is objectionable. It can be readily adulterated with barytes—barium oxide, BaO—or heavy spar, the specific gravity of which, 4.7, is so close to manganese, and its corresponding gray color will render the familiar commercial transaction easy of accomplishment.

Graphite, mixed with pure boiled linseed oil in which a small percentage of litharge, red lead, manganese, or other metallic salt has been added at the time of boiling to aid in the oxidation of the oil, forms a most effective paint for metallic surfaces, as well as for wood and fibrous substances. Wood surfaces protected by this paint, and exposed to the action of sea-water for a number of years, are found in a perfect state of preservation. Some recent experiments with this paint applied to the surface of boiler flues show that it prevents the formation of scale to a wonderful degree, and though it would seem in this case to also prevent the transmission of the heat to the water, yet it appears to be a better conductor than the scale which it prevents from forming. Steel tubes in locomotive and other high-pressure steam boilers pit more readily than iron tubes under the same conditions of service, so much so that on many railways the use of steel tubes is prohibited.

If the experiments now in progress with this paint applied to locomotive boiler tubes are as successful to prevent corrosion and the formation of scale as in the case of the boiler flues, its use will add materially to the economies of railway locomotive service, and can hardly be less effective in many cases on marine boilers.

In many places in the south-western section of our country the water is so bad in scale-producing qualities as to require the removal of the boiler tubes after only two weeks of service, or a little over a 2,000 miles run. Innumerable compounds of the anti-incrustation order have been tried and found useless to correct or even mitigate the trouble. If the simple application of a coat of plumbago paint furnishes a remedy for this evil, no better example of the "unexpected happenings" in favor of the engineer can be had.

The use of graphite for a pigment for the protection of iron from corrosion was inaugurated by the Joseph Dixon Crucible Co., and embraces an experience of over thirty years, during which period the use of it for the above purpose has given almost unqualified satisfaction in the many trying and test situations in

which it has been placed. Its use is increasing, and its merits are more generally conceded as being far superior in all respects to the oxide of iron paints for a protective coating on iron structures.

The amorphous graphite is usually mixed with more or less clay and silica, and being granular in character, even with close grinding does not give as good results as a pigment as the flake graphite, the grinding of which in raw linseed oil reduces it to a fine scaly condition; these scales overlap or felt together by the action of the brush in spreading, and constitute almost an ideal covering for resisting atmospheric influences. Its specific gravity is 1209, and the qualities used for pigment contain from 90% to 95% of carbon. The small amount of silica that accompanies it in its natural state is not detrimental to it as a pigment; in fact, ground silica is added to some brands of the paint, and its presence is thought to give better results than the absolutely pure graphite. It is practically unaffected by heat; temperatures under 600° F., the carbonizing point of the oil used with it, do not effect it; cold, dampness, acidulated mixtures, and alkalies have no affect upon it; and only the strongest of corrosive chemicals change it, and these only through the medium of the oil.

It spreads well over any surface to which it may be applied, and takes kindly to other paints, old or new, over which it may be placed, and as a priming coat for other colors it is almost unequalled. Its natural color is a slate, but the addition of light or tinting colors can be had without detriment to its preservative qualities. It is claimed for it that it dries quicker than Prince's metallic and other oxide of iron paints, without the use of dryer; that one pound of prepared paint will cover twice as much surface as any iron paint, or three times as much as any lead paint. A gallon of prepared paint weighs 10 pounds, and it will cover 600 to 700 square feet of iron work one coat, or about 550 square feet of wood surface.

It does not bleach or fade. Tin roofs covered with silica graphite paint last from fifteen to eighteen years without renewal, as against five to seven years of iron oxide paints. It is adulterated with soap-stone, pipe clay, and other like substances to the detriment of the quality of the paint. One pound of flake graphite ground with three pounds of soap-stone, neither the eye nor the touch can distinguish it from pure graphite, and there is no ready means of detecting the adulteration except in the laboratory.

An example of its use on a large scale is to be seen on the elevated line of the Pennsylvania Railroad at Jersey City.

Engineers requiring a ready rule to determine the amount of paint required to cover any structure, may find the following, from my own experience, to be of use.

Divide the number of square feet of surface by 200, the result will be the number of gallons of liquid paint required for two coats. Or, divide the square feet by 18, and the result is the number of pounds of pure white lead required to give three coats.

The tendency of iron to change its physical properties by a

FIG. 305.

change in the condition under which it may be placed in ordinary structural work, is strikingly shown by the following instance taken from *Engineering*, April 27, 1894; and reported by Mr. Oswald Brown, M.I.C.E., of 32 Victoria Street, Westminster.

"The cut shows portions of the bar dark and corroded, while the intermediate layers have remained bright. The bands of rust extend over both ends of the bar, giving it the appearance of being built up of layers of two different metals. The bar, which is of the best Yorkshire iron, gave under test the following results.

"Tensile strength 54,230 lbs. per square inch; elongation on 8 inches, 28.4%; contraction of area, 49.6%. No traces of lamina-

tion were shown during the test, but *some months after*, the bar was found in the condition illustrated, which seems to show that it really does consist of layers of slightly different chemical composition, those which have rusted being electro-positive to the other portions of the bar."

It is to be regretted that no analysis of the different layers of iron composing this bar is to be had. If the bar was not really homogeneous, the corrosion in the absence of any exciting cause must be attributed to the molecular changes due to the strain of the test.

If this is conceded, it goes far to clear up the mystery connected with the corrosion of steam-boilers, both internally and externally. Iron rivets and iron plates in some cases show the rivets corroded and the plates unaffected, and sometimes the contrary, and so with steel rivets and steel plates; also, iron rivets in steel plates or steel rivets in iron plates, all show the most erratic evidences, as regards corrosion; in many cases without reference to the character of the water used in the boiler, or to the external conditions. As a rule, all analyses of the plates, rivets, and other material used in boiler work are made from cold samples as they come from the manufacturers' hands, and before being worked; and when corrosion of either plate or rivet has commenced and attracted attention, it is seldom possible to get a sample of that particular make and lot of rivets to analyze to show what physical changes have taken place or developed in a nascent state, by the process of heating, closing the rivet, cold hammering the head, etc.

In no report of the corrosion of boilers that has ever come to my knowledge, has there ever been any mention of whether the rivets were hand-driven or power-riveted. The comparatively late advent of the power-rivet machine may account for this oversight. The writer's experience upon this point is, that hand-driven rivets corrode the soonest, and most rapidly when compared in the same class of work with power-driven rivets, and that the points of the hand-driven rivets are not so susceptible to corrosion as the heads, when exposed to the same corrosive influences. There is much available data in the ranks of the members of the A.S.M.E. upon this matter of corrosion of boilers in service, and under mechanical contortion from the effect of bad workmanship, such as riveting, flanging, connections of copper and brass, that are the foundation or starting point for the corrosion, particularly when

some slight and unnoticed impurity in the feed-water may act as the exciting cause. This data if brought out either by discussion at the annual or special meetings of the Society, would do much towards correcting the evil.

Free hydrochloric and free sulphuric acids are found in various sources of water supply, notably from canals and streams contaminated by mine waters. When these waters are used for steam purposes, any speck of carbon, manganese, or other impurities in the iron or steel plates, rivets, or pipe connections, act the part of the electro-negative element, around which the iron is rapidly corroded, pitting being thus produced. Soda ash made by the ammonia process is often used with good effect to correct the free acids in boiler waters. An ounce of phenol phthalein dissolved in a half-pint of methylated spirits and put in an ordinary sized boiler, will indicate by its light blood color that the water is practically free from acids or salts. When the color disappears, a solution of soda ash should be poured into the boiler until the blood-red color is restored.

Waters containing nitrates and nitrites in large quantities are common for sources of boiler supply. The application of carbonate of soda in solution to the boiler water does not always prevent corrosion and pitting, even when aided by a hard scale covering the boiler plates.

To counteract the effect of these nitrates and nitrites, metallic zinc is used, which decomposes them by combining with the oxygen and forming oxide of zinc, acting passively or as an electro-positive body to the iron, and converting the nitrogen into ammonia; also, neutral to iron. A mixture of one part by weight of bisulphite of soda to two parts of soda ash, also aids in preventing the action of the nitrates and nitrites on the iron; the tannate of soda also tends to prevent corrosion from these sources.

THE HASWELL ELECTRO-BROWNING PROCESS.

Another method or process for the protection of iron and steel has been recently invented by an Austrian, and is in practical use in Vienna. The United States patent No. 453,355 was issued June 2, 1891, and is described in the *Official Gazette*, vol. 55, page 1199. It is called the Haswell electro-browning or electro-enamel process, and consists in protecting the surface of iron and steel by

a deposit mainly of peroxide of lead, prepared by the electrolysis of a suitable lead salt in cold water.

The deposit is produced and completed in twenty minutes, and does not require more than one treatment. Several samples of it are presented for the examination of the members.

It appears to be best adapted to articles like shelf goods, hinges, and other cast-iron articles of general hardware, also axes, edge tools, scythes, etc., that require protection from changes of temperature that cause sweating, also from the effects of handling and storage in damp places.

The coating, while possessing considerable resisting power to the moisture, carbonic acid, and ammonia, that is in the air in its normal state; also, to diluted mineral and organic acids, and the chemical and other fumes of the workshop, is not, however, of so high resisting power as to withstand immersion in either water, sea-water, or acidulated mixtures beyond forty-eight hours, which under these conditions appears to be about the limit of its protective qualities. It can be polished on ordinary buffing wheels after it is applied, as shown by the round samples, it being unnecessary to polish the article before browning it, its color then being decidedly pleasing, while the coating on the rough ground surface of the sample is far from objectionable.

The sheet-iron sample shows its effect upon iron as it left the rolls, a pickling to remove the scale being the only preparation required. The sheet-iron can be bent to a right angle over a moderately sharp corner without cracking the coating or injury to its protective power.

The coating has no chemical affinity with the metal which it is to protect.

Whether dry or moistened it creates no changes on the surface of the article coated, and is not affected by mechanical injury like a blow, so as to flake off, expose the surface, and cause a rust spot to form.

It stands hard rubbing with pumice stone wet, or dry, on a piece of felt, is not affected by the usual changes of temperature of the atmosphere, and bears a temperature of 350° Fahr. without injury.

The cost of material in the electro bath is ten cents for one hundred square feet of surface coated. The apparatus required consists of a few tanks to hold the lead solution; a few reels or racks to support the articles, according to what they are; the whole

at a cost of from $25 to $250. An ordinary electroplating dynamo is required, the size depending upon the amount of work to be done, and power to drive the dynamo, etc., from 1½ to 2 voltage is all that is necessary.

Water must be provided, and drying facilities by steam oven or otherwise as is most convenient. The articles must be cleansed from oil or finger marks before immersion in the bath, but a limited amount of skilled labor is required in the process.

The royalty for the use of the patent is not included in the above statement of cost. The technical value of the invention as to cost of application and its effectiveness, appears to rank next to the nitre process of Lieut.-Col. Buffington heretofore described; but for all purposes requiring the preservation of the finish and cutting qualities of highly tempered edged tools, it has no competitor.

The coating is easily removed by friction with turpentine, concentrated lye or benzine.

Its application to finished shafting and kindred articles is feasible and practical, but on pulleys, machine tools, engines, and like bulky articles, the old white lead and tallow pot will hold its own for yet a while, and the machine-shop apprentice will not be deprived of a chance to develop his artistic proclivities.

There are a few compounds for the temporary protection, rather than the preservation, of machined or bright surfaces of iron and steel, that, not as enduring as paint, may be denominated lotions. Some of them are effective and easily applied.

Krupp uses a compound called Mannocitin for his bright steel, machined articles, and ordnance in store. It does not turn rancid or harden with age, is consistent enough to resist torrid temperatures without slumping off, is easily removed by turpentine or benzine, and is a valuable substitute for the old tallow pot product.

Coal-tar, a product from the distillation of gas coal, when properly prepared, forms an effective protection from corrosion of iron and steel from ordinary atmospheric attack, and if reduced to a pitch compound and applied to the metal in form of a hot bath, in which the article is immersed for a sufficient time to allow it to attain the same temperature as the bath, the coating formed is but slightly inferior to red oxide of lead paint, and in some applications is much cheaper.

In all cases the removal of the scale of the rolls or foundry sand

produces superior results. The coating formed has about the same elasticity as the metal, and withstands mechanical injury very well. The composition of the tar has much to do with the success of the application.

Coal-tar from gas coal and cannel coal, as it is collected from the hydraulic main, by analysis consists of the following elements as averaged from a large number of samples. A 40-gallon barrel consisting of:

1½ gallons, or 3.75%, of light oils, consisting of benzole, naphtha and carbolic acid.
9½ gallons, or 23.75%, of heavy oil, consisting of creosote oil and anthracine.
29 gallons, or 72.5%, pitch.

Boiled in open kettles, this should be reduced from 15% to 25%, according to the duty required of it. The tar resulting from the distillation of petroleum oils for gas is of a decidedly inferior quality to that obtained from gas coals, and is better adapted for coating the cruder forms of wood constructions, piles, dock timbers, fence posts in the ground part, than metal work. But this same oil tar, if distilled at heats from 600° to 800° Fahr., forms a pitch of almost adamantine hardness when cold, and resists almost all corrosive agents and solvents except those of the hydro-carbon class.

When the concentration of the gas coal-tar is carried to the 25% or 30% stage, the product is comparatively odorless, or at least is not any more objectionable than that from oil paint, the pungency due to the light oils and carbolic acid being dissipated.

GALVANIZING PROCESSES.

Galvanizing or coating metals with zinc, and its alloys of tin, antimony, arsenic, lead, etc., is a well-known process, and its value is well recognized, as the extensive application of it for a large variety of purposes shows; but it is easily attacked by weak acidulated liquids, sulphurous and other atmospheric fumes, and the protection afforded, except in fairly protected and favorable locations, is but temporary. Its mission is so short-lived, and corrosion once commenced proceeds so rapidly in unlooked-for spots, that one is forced to condemn its use except for locations of minor importance, or where its failure as a protective agent will not be attended with serious results. The Ajax patent coating, so extensively advertised a few years

ago, was a coating of lead and its alloys applied to iron surfaces which had undergone a peculiar pickling and cleaning process to free them from scale, sand, etc. Other than this point it had no merit over the ordinary galvanizing process. Zinc and tin, with the other before-named metals, are by themselves all electro-positive to iron and steel, but when alloyed or combined with each other are more potent to induce electrolytic action upon the baser metal and hasten decay.

Galvanized iron is one of the worst materials to hold oil or paint. Out of a large number of the best quality paints applied to galvanized iron roofs and cornices, but one, a patented paint of an undetermined nature, adhered with any marked degree of success; with ordinary trade colors, this difficulty to take paint is greatly increased. The cause of this is not known, but a plausible theory is, that the sal-ammoniac used during the process of galvanizing forms a film of basic chloride of zinc on the surface of the metal during the process, which being more or less hygroscopic, repels or prevents the close adherence of the paint to the surface. Zinc in all forms as a pigment has a peculiar action upon all metallic or other surfaces with which it is brought into contact as a coating. Zinc white paint *peels* from all surfaces coated with it. This action, no doubt, arises from the zinc combining with the oil to form a species of zinc soap, which gradually becomes a hard, brittle, non-adhesive substance, easily affected or washed away by water or watery vapor.

The selection of the galvanizing with zinc process for the protection of the wire used in the main cables of suspension bridges, is possibly a mistake of the most serious character, and the adoption of this method of protection from corrosion by some of the most eminent engineers in the world for their most important and world-renowned structures, is either an example of gross ignorance of the subject of the causes of corrosion in iron and steel and the methods of protection from its effect, or else is an utter disregard of the importance of those little things, or next to nothings, that in many cases have turned what primarily was a success into an ultimate and unpardonable failure.

The main cables of the Niagara Suspension Railway Bridge, only thirty years after its erection, were found to be so badly corroded that the strength of the structure was seriously affected, the factor of safety having been reduced nearly thirty per cent. by the corrosion of the wires, with the possibility in plain sight that

the structure would not stand another like period of time unless repaired and restored to its original condition.

The preparation of these wires to resist corrosion was: before they were strung into the cables, they were galvanized with zinc, then they were coated with one or more coats of boiled linseed oil with Japan dryer, and when the requisite number of wires had been strung to form the cable, the cable was bound with a close-wound wrapping of wire similarly galvanized and oil-coated, as an outside protective covering. Then the whole cable covering, iron straps, fastenings, saddles, and all the connecting parts, were given a number of coats of white lead paint, and the job was declared complete and able to defy for ages the elements of corrosion or any other destructive agents of iron structures, not even excepting excessive strains from overloading or those changes supposed to exist in the form of crystallization due to vibratory impulses, polarity of position, etc. All of the work was apparently well planned, and executed from the best of material, regardless of expense; the reputation of the engineer corps engaged in its construction was much prated about and carefully considered at all times; and any causes likely to affect either the strength of the bridge or the reputation of its designer were supposed to be eradicated in this, at that time considered the highest example of engineering in the world.

As a further means of protection, when the cables thus wrapped, painted, and—hollow mockery of words—protected, entered the ground, or anchorage pits, they were bedded to a greater or less extent in good hydraulic cement mortar, which, from its nature of quick setting, *was supposed to be dry*, and so it was, so far as the term dry is used to express the character of any masonry material buried in the earth and exposed to the ordinary conditions of earthy contact; but as compared with the character of the same object exposed freely to atmospheric effects, it was perpetually damp, *if not wet*, and always in a prime condition to inaugurate and perpetuate the process of corrosion of any iron body in contact with it. This condition was aided by the inclination or pitch of the cable from the horizontal; any condensed moisture or water that could find a crevice in the outer wrapping of wire and paint necessarily followed natural laws by seeking its reservoir at the lowest points, and it was near these points that renewals of the corroded wires had become necessary, and were made by cutting them out and replacing them with new

ones, and the removal, so far as was possible, of the contact masonry and cement, and allowing the cables access to the air, and where they could be readily inspected at all times to ascertain their condition. So far well, but the condition of the wires upon the inside of the cables between the suspension towers is another matter, and whether corrosion is not here in active progress is a question not easily answered; neither can inspection at these points be had with anything near the facility that the anchorages gave.

These wires, during the period of stringing and grouping them into cables, a process occupying months of time, were exposed to all the atmospheric changes, including many storms of rain and frost that must have soaked the freshly oiled surfaces of the wires and filled the interstices between them so thoroughly that no ordinary conditions of drying would have evaporated the water thus collected, and when the close-wound wire wrapping and the paint was in place all the contained moisture was practically as well sealed up for the attack upon the iron as any plan could have been devised to retain its fellowship with the metal. Even the relief that our masonry engineers receive from the presence of *weepers* to free them from their tribulations, from the nature of the case was denied to the bridge-builder.

The difficulties that attend the application of any paint to a galvanized surface are referred to elsewhere in this article. That it is in *this case a wire*, does not obviate the trouble, neither did the application of the boiled oil coating afford but a limited protection during the period of erecting the bridge. Freshly dried oil absorbs moisture as readily as a sponge; the wire cable wrapping prevented any circulation of air to dry out the moisture that from any cause found its way into the cable beyond the wrapping; also deprived the oil of free access to the air to obtain the oxygen necessary for changing it from its hygroscopic character to that of a repellant one that only a long time can give. These mistakes cannot now be remedied, but naturally ought to be avoided in the future. Had it been proposed to wrap each wire in a dampened sweat cloth, and then to bind them into a cable as a *part of the method for the preservation of it from corrosion*, the proposition would have met the jeers of the veriest engineering tyro in the land. How far the method actually adopted is removed from the sweat cloth, will not want a much wiser head than the said tyro's to answer.

In the case of the Niagara Bridge the last of the preservative methods employed, viz., the external coating with white lead paint, was probably the best of all. When soiled and dingy from exposure, it called for its renewal more forcibly than some of the stronger and better protective paints; and in order to please the eye, its adoption was admissible; but its mantle, like charity, covers a multitude of (engineering) sins.

The old game "follow your leader" appears to be not yet forgotten, as the Brooklyn Suspension Bridge is protected from corrosion by almost the identical methods employed at Niagara, and which have been *found inadequate in all respects to protect;* the principal exception of these methods on the Brooklyn Bridge being at the anchorages, where the wires were replaced by eye-bar connections which are exposed to the air for the greater part of their length.

It is also a point for consideration in connection with both of these engineering structures, what effect the method of attaching the small cables that support the roadway superstructure and connect it to the main cables has upon the prospective life of these bridges, and whether they will not prove the most serious of the causes to hasten corrosion and the ultimate failure of the structures after a comparatively short life.

These main cable bands are made from iron of comparatively heavy sections, approximating $5 \times \frac{5}{8}$ and $\frac{3}{4}$ inches, and were forged as open-jawed hoops on formers representing the outer and extreme size of the cables over the wire-wound wrappings, and then they were opened out so that the jaws of the hoop would pass over the cables; and when applied to the cable they were brought to a bright red or working heat, sprung over the cable, closed down by a bolt through the jaws, and swedged by hammer and fuller to place and a close fit on the wire wrapping.

How well a film of dried linseed oil can withstand such treatment and still be an effective coating, I am loath to say; or what effect such a mechanical application of heat and sledging would have upon the wire wrapping or the cable wires inside, that have had a previous preliminary dose of the screw band and sledge to get them into the circular form preliminary to wrapping. It requires but a very slight injury to a galvanized iron object to start corrosion, which once begun proceeds apace; rust begets rust, and when it is considered that these bands as danger points on either of the said bridges are hundreds in number, and no dabbing

of the paint-brush can reach any part of the surfaces covered by these suspension hoops, the future successor in the trust of caring for these bridges and others of like construction has no enviable position, and will require to employ more effective methods of inspection than to speer at them through a spy-glass from a mile or more of distance.

The tendency of iron oxide (iron rust) paints to burn out or decompose the oil with which they are mixed, harden with age, then scale off, was well known for years prior to the construction of either of said suspension bridges.

The United States Navy Yards, the Dutch Dock Board of Construction, the British and French Admiralty Bureaus, have recorded many experiments and tests of the qualities of iron

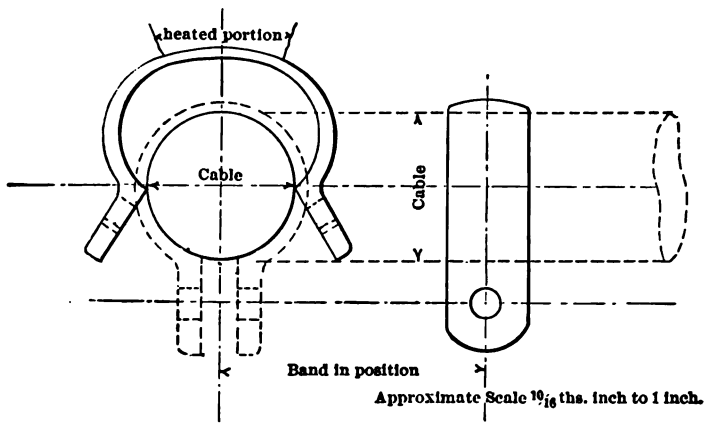

FIG. 306.

oxide paints to prevent corrosion; the decisions of the said bureaus are against the use of any oxide of iron paints upon any iron or steel work of magnitude or importance. Notwithstanding this available experience, the entire iron superstructure of these bridges and the small suspension cables connecting the superstructure to the main cables, were primarily painted with iron oxide paints before erection, and again when in place; the wrought-iron work was not even cleaned to remove the scale of manufacture, so that the paint, poor as it is for protective purposes, could get its proper bond upon the metal it was spread over.

The Brooklyn Bridge is exceptionally exposed to corrosion, having not only the ordinary atmospheric conditions, but those

due to the sea-air; also the decomposition of animal matter from the drippings of the driveway. There is a large area of surfaces in the members of the superstructure and its connections that are comparatively covered from the free circulation of air, that would ordinarily dry out the moisture that reached them in the form of rain, snow, frost, fog, and the animal drippings. It is the tendency of these agents to collect upon the horizontal parts of the structure, and to concentrate their action in spots in the form of drops, where, as a rule, corrosion will always be found in progress to a greater or less degree, and which should have been particularly guarded against.

The difference in the cost of the application of well-known and well-proved methods for the prevention of corrosion, over those adopted in the Brooklyn Bridge, could not have exceeded one-half of one per cent. on the cost of the entire structure; and in this case, where the cost amounted to many millions of dollars, and the exigencies of the traffic over the bridge preclude closing it for repairs (as a country road toll-gate would drop the bars and send the tide of travel some other way), it seems to be a case of woeful oversight that better protective methods were not employed to preserve the structure.

The covering of church spires and court-house cupolas with bright tin used to be a familiar method of marking the importance of the structure to the rustic communities. These were undimmed by paint, and in many cases were uncorroded and bright after fifty years or more of exposure; but the coveringwas pure tin plate and honest work, and when leaks did occur in the covering it was always in the seams where the solder had introduced some lead alloy to begin its incipient work of decay.

When the tinning fraternity began the use of acid in connection with resin (and in many cases used acid alone) as a flux to the solder, the life of the roofing was materially shortened; the job hastened to completion, but the tinker never took a cent off the bill.

The advent of coal as a domestic fuel instead of wood, as well as the fumes of coal smoke from multiplying manufacturing establishments, soon produced an effect upon the bright tin roofs, and paint of varying qualities, and in too many cases of a cheap John order, as being good enough for a house roof, became the order of the day to preserve the work. At present the fierce competition for orders between our free trade brothers across the sea, as well

as the unscrupulous character of some of the Budensiek contractors and builders—and in many cases the engineering craft are guilty of acquiescing in and calling for cheaper materials—has brought into the market a quality of tin and terne plates that have about reached the maximum point of cheapness and nastiness as a vehicle to hold paint on a metallic surface. As an engineer remarked when questioned as to the quality and merits of a special brand of tin roofing plate, imported for a specially fine job: "Oh, yes, we had it painted when it was boxed and shipped, it was painted again when we received it and took it out of the box, painted again when it was ready to lay in place, and after laid had two coats given it, and the business lasted about six years, when we ripped the thing off in midwinter and laid a tarred felt, coal-tar and gravel roof, which has been down twelve years, and is sound and tight as when laid, and did not cost one-half as much as the blessed tin did, including the expense of taking the thing off."

The failure was due wholly to corrosion from the inside outward. Certainly all of the plates, nearly a thousand in number, could not all have been improperly pickled, scaled and cleansed before coating; but they did receive from the tin bath a uniform application of a tin and lead or other alloy coating, that pitted in a dozen places almost every plate laid on the roof. I hardly think it advisable for any one to approach that engineer for an order for tin roofing plates even if backed by a strong guarantee.

The alloy of zinc, lead, antimony, arsenic, etc., with the tin used in the cans and vessels in which so large a quantity of our daily food is now marketed, is a serious menace to the public health. The most casual inspection of "the dump" that graces all our cities, towns, villages, hamlets, and even the solitary cottage beside the sea or 'neath the sheltering hill, cannot but convince one of the dangers we are daily exposed to from the cumulative poisons taken into our systems from this cause, and though we as a nation are wont to demur to the too close inspection of and interference in our so-called private affairs, matters are so fast assuming a crisis stage from our greed for the almighty dollar, with the sentiment as a ruling passion: "No matter what happens so it does not happen to me," that relief and reform must be had, even if national, state, city, and town inspectors and boards of health one and all are called into being (happily if they could be divested of all political character) for the purpose of curtailing and controlling this growing cause of disease and death.

The terror that lurks in the deadly tomato can is fast losing its joke and becoming a sober reality. The benefits that arise from a careful inspection of the metal wares enclosing food products, even if the proceeding may partake too much of the paternal government character, is strikingly shown by the examination of the French articles so prepared, and a comparison of that tinware with the best samples of English, German, and other Continental countries, and noting its superior qualities, and how remarkably they resist corrosion or change from any of the vegetable acids in food or that due to fermentation. Our American tin plate manufacturers have much to learn in this matter, and might study French methods of manufacture with better results than those the imported Welshmen are striving to introduce.

Mr. Asquith, the Home Secretary, has certified to the English Parliament that the processes of tinning and enamelling hollow-ware, cooking utensils, and the canning for food products, are dangerous and injurious to the public health, and asks for legislation to correct the evil.

The German Government has lately issued instructions to the inspectors of food products, that in all domestic cooking utensils and metallic cans for containing food the percentage of lead shall not exceed 17% to 18% of the tin in any case.

An alloy of three parts lead to one part of tin is of frequent occurrence in American domestic tinware (imported plate). Vinegar readily dissolves out the tin in this case, leaving the lead, which is quickly acted upon and dissolved by hot water.

In a cup of tea or coffee drawn from water heated in a vessel of this character, the lead is readily detected as being present, and as it acts as a cumulative poison the danger in the use of such ware is apparent.

An alloy of three parts lead and five parts tin is frequently used to tin copper cooking utensils. These proportions contain two times the amount of lead per unit of tin that the German standard restricts the use of.

One of the most trying situations for the protection of his work that the engineer has to deal with, is in the exposure of wrought or cast iron water and gas pipes, through the cinder beds or fillings from blast furnaces, rolling mills, etc. Pitch compounds applied hot, afford some measure of protection over that of the naked iron, but the creep of the pipes due to changes in the temperature of the fluids passing through them, and of the surrounding earth,

the concentrated action due to the sulphur in the cinder or ashes, the porous nature of the covering over the pipe and in which it is embedded allowing the ready access of air and moisture from the streets, ammonia and other fumes and liquids due to the decomposition of a score or more of different bodies, all aided by the vibration caused by the passage of heavy trucks, railway trains, and kindred causes, combine to render the problem one not easy of solution.

A thick puddle of good clay of a thickness all around of at least one-half the diameter of the pipe (and in case of pipes under eight or ten inches diameter, the thickness should be equal to the diameter of the pipe), if allowed to dry out well before covering in, gives a good protection, and aids materially whatever compound is applied externally to the pipe. I do not know of any case where the pipes coated with magnetic oxide by the Bower-Barff or any other process has been tried; but it seems to the writer that if there was any situation in which the merit of the process could be demonstrated, this is one, and it is not one of minor extent or importance.

The rapid destruction of the wrought and cast iron mains, as well as the galvanized iron and lead service pipes of our water and gas distribution systems, by electrolysis brought into action by the use of the system of pipes as a ready ground connection for our extended and fast-multiplying lines of electrical service, is assuming so serious an aspect, particularly upon streets or locations where electrical railways are in use, as to call for serious consideration by all branches of the engineering fraternity for some method of protection other than now in use.

So rapidly is the destruction progressing, that in many places the utter prostration of the water and gas systems may be looked for within the next decade. Pipes now affected, removed and replaced with new, but pass the corrosion further along to other and weaker points in the line, until the whole system is becoming honeycombed with points ready for failure on a little change in the ordinary working conditions.

The return current from the single wire trolley system of electrical car lines appears to seek out and take for its passage back to the power-house the pipe system contiguous to the line, instead of using the ground wire connections provided for it. The use of the electric motor now contributing so much to the comfort and conveniences of life in the form of electrically driven ventilators,

pumps, elevators, and other small powers, is rapidly extending the evil. The remedy, if not provided by the electric light, heat and power companies in the form of taking better or sole care of their product from its generation to its extinguishment, will ere long be found in mulcting them for damages so heavy that the small margin now existing between the dividend point and a receiver will be wiped out.

Some experiments on a small scale of the Bower-Barff magnetic oxide application to pipes and iron exposed to electrolysis action, appear to indicate its ample protective power. What it would develop upon an extended system of water or gas service remains to be tried. The iron pipes and connections, if protected by this process, would still leave the lead joints and lead service pipes exposed to attack. Cement joints might obviate part of the evil, but would render the main pipe line too rigid in

FIG. 307.

withstanding changes of temperature, and expansion joints are not to be considered in pipe lines of hundreds of miles in extent.

The electrical engineers as a body appear to have given this question only a sort of a happy-go-lucky, or after-me-the-deluge consideration; but the hour is not far distant when they will be walking the floor crying, "What shall we do?" instead of their water and gas brother who is now having his inning.

Through the courtesy of Mr. Ralph W. Pope, Secretary of the American Society of Electrical Engineers, I present a number of cuts showing the effect of electrolytic action upon the underground water and gas systems of the country.

These cuts illustrate a paper presented at the April meetings of the American Society of Electrical Engineers, New York and Chicago, by Mr. Isaac H. Farnum, E.E., and reported in

FIG. 308.

FIG. 309.

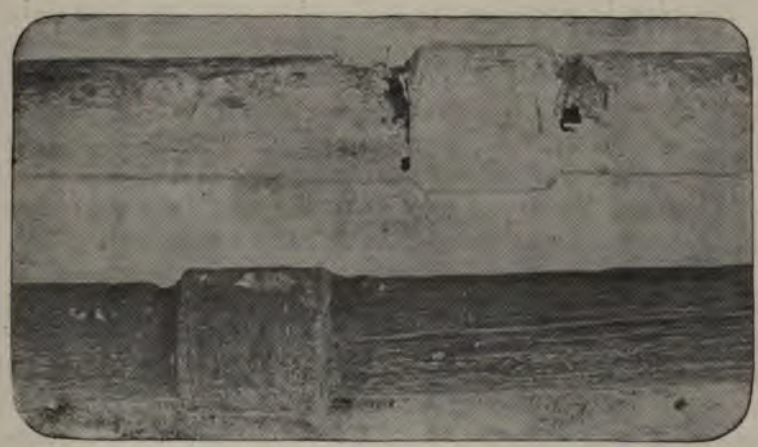

FIG. 310.

the April number Transactions American Society of Electrical Engineers, Vol. XI., No. 4. This paper was discussed at the May meetings of the said society at New York and Chicago.

See May, No. 5, Vol. XI., Transactions American Society of Electrical Engineers.

Mr. Farnum's paper and the discussions upon it bring most of the accurate and available knowledge of the question down to the present date, and can hardly fail to interest the members of the American Society of Mechanical Engineers.

Fig. 307 shows the corrosion of a small iron service pipe from the Cambridge (Mass.) gas system.

Fig. 308 is a 6-inch cast-iron water main from Milwaukee, Wis.

Fig. 309 is a service pipe from Albany, N. Y.

Fig. 310 is an iron gas service pipe from Brooklyn, N. Y.

The protective power of lime-mortar and hydraulic cement to prevent corrosion in iron and steel under certain conditions has been demonstrated by centuries of actual use; yet the facts thus established are too often ignored by the engineer of to-day.

We do not have to turn back many pages of the record book of success and failure, to find a remarkable instance of the latter, in the utter collapse of the Bayliss patent wrought-iron water pipe, that a few years ago was brought out as an indestructible, imperishable article with a hygienic annex, to take the place of cast-iron pipe for water distribution.

The pipe in question was made of wrought iron from $\frac{1}{16}$ inch to $\frac{1}{8}$ or more inch in thickness, according to the size and pressure to which it was to be subjected; formed and riveted together in the shop or worked to shape upon the ground from the sheets, by a portable lot of rolls, punches and other tools needed, into sections about five feet or more in length; these sections being tapering, and if shipped were packed in nest form conveniently and cheaply so far as transportation was concerned.

The tapering sections were laid in the pipe trench and lined *in situ* with hydraulic mortar, the large end of the section forming the bell for the small or spigot end, the joint being made with the cement, and the outside of the pipe was also covered with the cement mortar, to the depth of an inch or more, all as far as possible at one operation, depending somewhat upon the size of the pipe, the location and the general conditions of the work, setting of the mortar, etc.

The advantages claimed were a continuous pipe from end to end, no matter what its size or length, of uniform strength, lined with a non-corrosive and protective, tasteless and innoxious cem-

ent, and guarded from external injury and corrosion from acids in the soil in which it was laid, and atmospheric influences, by the same or a similar cement covering.

The projectors wholly ignored the difference in the co-efficients of elasticity between the iron and cement, depending upon the integrity of the lining to protect the iron from the water; this lining once broken admitted the water pressure to the iron, the thin riveted joints of which it was impossible to calk tight; the pressure thus reaching the external coating, which was unable to resist but a small head, cracked and flaked it off, and the destruction of the whole line of pipe followed rapidly, and as a matter of course, after but a short period of service, most if not all of the lines being abandoned and re-laid with cast-iron pipes inside of ten years.

In general no attempt was made to coat the iron with zinc or any coating, or to remove the scale of manufacture. When the pipe was thrown out of the trench finally and broken up, the outside coating came away, bringing with it the scale and oxidation under it as a porous plaster on removal takes away all over which it was spread. The connection of the house-service pipes with the main line gave no end of trouble to make and keep tight, and the whole system was a forcible illustration of the importance of those "next to nothings" that so often escape the attention of the best engineers, and bring no end of disappointment to those who are financially responsible for the introduction of the new idea.

ELECTRO-PLATING WITH COPPER AND ALUMINUM.

Possibly the most extended application for a single purpose of the use of any process other than paint for the preservation of iron and steel exposed to atmospheric attack, has been made at Philadelphia, Pa., in the construction of the iron-work of the tower of the new City Hall, now under process of erection and nearly completed. I am indebted to Mr. F. Schumann, C. E., President of the Tacony Iron and Metal Co., Tacony, Pa., contractor for the work, for the following data. The *Scientific American* also describes and illustrates the process in the following issues, the data for which was also furnished by Mr. Schumann.

October 22, 1892, Vol. 67, No. 17. Aluminum Electro-Plating in Architecture.

September 9, 1893, Vol. 69, No. 11. Electro-Plating with Copper.

This process is a double one. The first one is designed to protect the iron from rust by an electro-plating of copper of 14 ounces per square foot of surface, and a finishing coat of an alloy of aluminum and tin of 2½ ounces per square foot, for color to harmonize with the stone-work of the lower stories of the tower; also to prevent oxidation of the copper into a green coating of verdigris.

The process was adopted as a substitute for paint, the periodical renewals of which would cost $10,000 per annum, the primary coat of which was to be boiling linseed oil, in which all material was to be immersed until they had attained the temperature of the bath. The total weight of wrought and cast iron to be protected is about 500 tons, and comprises 100,000 square feet of surface, the largest single pieces being 16 columns 27 feet long and 3 feet in diameter, weighing 10,000 lbs. each. These columns received the copper coating inside as well as outside. The outside coating, being most exposed, is a double one, requiring two operations or baths, while the aluminum coat is given last as the protective coating to all beneath it. The cost of the whole process varies from 40 cents per square foot to $1.00, depending upon the shape of the piece—simple plates, rods, angles, etc., costing the least, while curved pieces, with large lugs, flanges, with core holes, cost the most, the principal expense being the cleaning, the *greatest care* being necessary to insure good work.

The capacity of the Tacony plant is about 250 square feet per 24 hours of from 12 to 16 ounce per square foot coating; though some recent improvements by Mr. J. D. Darling, manager of the plating works, have nearly doubled the capacity of the plant, and avoids the trouble from "spit holes," blow holes and stodgy places, that give a great deal of trouble to search out and *solder up*. The electro-plating covering, in these cases, evidently follows the peculiarities of galvanizing, where the smallest needle-hole will be coated, but will not fill up the crevice.

The manipulation to apply the process necessarily varies with the shape and size of the piece. Suitable cranes, trolleys, hoisting gear, trucks, etc., being provided to handle one of the large columns referred to, the operation of coating it is as follows:

The column is first immersed in a tank containing a strong solution of caustic soda, heated by a steam coil, and is boiled a number of hours to remove all the grease and oil due to the machining process. It is then removed and thoroughly washed in water from

a hose, then placed in a second tank and pickled with diluted sulphuric acid until all rust and scale are dissolved and loosened, then thoroughly cleaned with steel brushes and water, and goes to the third tank containing the cyanide plating solution, and receives its first coating of copper. It is then removed from the bath and transferred to another tank, and given a coating of paraffine wax *inside;* thence to the fourth tank containing an ordinary copper-plating solution, where it receives a heavy coating of copper, about sixteen ounces to the square foot of surface. The paraffine is then boiled off, and the column enters the sixth or aluminum tank and receives a heavy deposit of aluminum, two to three ounces per square foot. It is then well washed with pure water and is ready for erection.

The columns and other pieces are brought into the electric circuit by wires passing around them like slings, and attached to a conducting brass bar over the tank.

In the cyanide tank a current density of three amperes per square foot is employed; in the acid tank, ten amperes; and in the aluminum tank, eight amperes. Separate dynamos for the several tanks are necessary, the developments being 1,000 amperes at six volts, 2,000 amperes at eight volts, and 4,000 at two and a half volts.

Mr. Darling's improvements in the acid solutions enable copper to be deposited at the rate of twenty to twenty-five ounces per square foot in twenty-four hours, that is as malleable and almost as smooth as rolled copper; and copper deposits one-eighth of an inch thick, or ten pounds per square foot, have been made in twenty-four hours.

For rolled sheet-iron and steel, and in rods and shapes where the surface is free from sand-holes, eight to ten ounces of copper per square foot of surface will be sufficient, while on rough cast-iron or steel work fourteen to sixteen ounces will be required.

These amounts are greatly in excess of those given in books treating on electro-plating, but practical experience shows that the above amounts are necessary to give a protection that will prevent any appearance of rust, and will last as long as the structure will stand.

For inside work, when the plating is used more for effect than as a protection, zinc, tin or aluminum is added to the copper bath to give a bronze appearance, and a coat of two or three ounces to the square foot will suffice.

The new Bourse building, to be erected in Philadelphia, will have examples of both kinds of plating. The outside iron-work, window-frames, etc., will have a heavy protective coat of copper, while the inside work will be electro-bronzed.

In France, where copper-plating to protect iron was first used, the copper was not deposited directly on the iron, but on a coating of varnish, rendered conducting with plumbago or powdered copper, applied to the iron surface and allowed to dry. This was done to avoid the difficulties of cleaning the cast-iron and the use of two solutions in depositing the copper; as the article, after the varnish was dry and the plumbago applied, was placed directly in the acid solution.

This method gives a coating that is not firmly attached, and is liable to be torn off by mechanical injury. The lamp-posts of Paris and the beautiful fountains of the Place de la Concorde and the Place Louvois are examples of this method.

The method used in America deposits the copper directly on the iron, and the article so plated may be bent and twisted into any shape without detaching the copper deposit. As it is often asserted that aluminum cannot be deposited from an aqueous solution, the following information, furnished by Mr. Darling, may prove of interest:

"Although aluminum is generally credited with indestructible qualities, and high resistance to corrosion, it has but few qualities that would make it advantageous as an electro-deposit upon other metals; for while in a massive state it resists atmospheric action and retains a certain brightness for a long time, when it is deposited electrically from an aqueous solution, which deposit is of necessity of a more or less porous nature, it soon tarnishes and assumes a dull, bluish-white color, when exposed to the direct action of the elements. But for a protective coat, say for copper, for which purpose it is used on the City Hall tower, it answers very well, as the slight superficial oxidation that takes place, protects the metal underneath from further attack, and the neutral color that it assumes harmonizes well with the stone-work of the tower.

"Aluminum is no doubt more difficult to deposit than any other of the common metals. This is because of the high voltage necessary to decompose aqueous aluminum solutions, and the tendency to re-dissolve after being deposited. We have not got the thermal data required to calculate the potential difference or electromotive force necessary to decompose the different aqueous solu-

tions of aluminum, but reasoning from analogy, it must be several volts in each case, and as water requires only a minimum electromotive force of 1.5 volts to decompose it, it would seem at first glance that a compound which requires over two volts for its decomposition in aqueous solution would involve the decomposition of the water, and therefore would be impossible. But in reality this is not so, as may be seen in the case of the caustic soda, which requires over two volts. Yet sodium may be obtained by its electrosis if mercury be present to absorb it and protect it from the water. The fact is, that when two substances are present requiring different E. M. F. to compose them, if the E. M. F. is high enough to decompose the higher compound, the current is divided between them in some ratio decomposing them both, and I find by using a solution of aluminum that has but a slight dissolving effect on aluminum, with a density of current of 8 amperes to the square foot, with a sufficient high voltage (6½ to 7), aluminum can be deposited on the cathode at the rate of one gramme per hour per square foot, in a reguline state, and with higher currents it can be deposited much quicker, but it will be in a pulverulent state, which does not adhere."

Recent improvements in this process form the basis of a patent for a proposed use of electro-copper plating for the protection of ships' bottoms, applied quickly and cheaply in sections, by means of movable and portable baths, the deposits overlapping each other, the entire hull being thus treated during construction, or it may be applied to vessels already built. How effectual this application for the purpose named may prove, is a matter of some doubt, owing to the difficulty of thoroughly cleaning the surfaces to be coated, and which appears to be an indispensable condition for success.

The irregular contour of the immersed parts of the ship being plated *in situ*, as well as the porous nature of the copper deposit requiring a heavier coating than is absolutely necessary for protection, and whether some insulating or protective compound interposed between the copper and the metal of the ship will be required to prevent electrical action, all are questions to be settled by the slow test of time, which does so much to upset the best-laid plans of mice and men.

If effectual, the cost of application at even the highest price named by the Tacony people, $1 per square foot of surface, would be small hindrance to its general adoption, it requiring

only about $80,000 to coat our largest war vessel on its inside surfaces as well as the outside surfaces, joints in the armor, and all places not of ready access for the application of paint compounds.

JAPANESE LACQUER.

The adaptableness of this natural vegetable product to the preservation of metallic surfaces, as well as those of wood, paper, and other fibrous bodies, has never received the attention of engineers that the industrial importance of this method of coating and protecting surfaces demands. The general idea that its application is one of art, and is only adaptable to bric-a-brac, is wholly erroneous. The Japanese use it for an infinite variety of purposes—acid tanks, coating the keels of ships, highly finished coach and decorative panels, and articles for domestic use, resisting hot water, soap, and alkaline solutions. It may be truly said that were it not for the bamboo and lacquer trees, life for the Japanese would hardly be worth living. There is no reason why the lacquer tree should not thrive in this country. Its sap, which is used as the material for all lacquer work, is a natural essence, and vastly superior to any varnishes used here. Unlike even copal, which is an artificial mixture of resin, fatty oils and turpentine, Japanese lacquer is a ready-made product of nature that hardens into a mirror-like smoothness, never splits nor cracks, and is of great durability. The art-lacquer work of Japan is essentially individual, and ought not to be treated as an undistinguishable whole. The superiority of the art work is due not only to the special merit of the material, but also to the care and skill shown by the Japanese in the manipulation of it.

There is as wide a distinction between the ordinary lacquer tray or cabinet of commerce, and the exquisite lacs of the great Japanese artists, as between a theatrical poster and a canvas of Raphael. Each of the great masters of lacquer has created a style of his own, and has founded a school of which the traditions have been kept alive by his successors for centuries. At the Centennial Exhibition of 1876, in the Japanese department, there were exhibited plates, pans, etc., evidently of a common quality of lacquer used by sailors and others for domestic purposes, also some samples of a finer grade in trays and cabinet work, that

had been sunk in the sea over fifty years, and though covered with barnacles and other marine growths were practically unharmed, so perfectly had they been protected by the lacquer. It seems almost incredible that so valuable an article that can be produced as cheaply as maple sap has remained comparatively unknown in this country and Europe, at least so far as applying it to the palpable protective purpose that nature evidently designed it to be used for.

Mr. Hotta, a lacquer manufacturer at Tokio, has applied a preparation of lacquer to the hulls of the iron and steel war vessels lately purchased by the Japanese Government. The applications made in 1886, and subject to frequent inspections, appear to be of the most encouraging nature so far as preventing corrosion. The attempts to meet all the requirements in the case by making the lacquer anti-fouling as well as anti-corrosive, also appear to be eminently successful, as the coatings applied in 1890 show no signs of either fouling or corrosion. The first applications of an anti-fouling paint *applied over* the lacquer were failures, the urushic acid of the lacquer attacking the metallic base of the anti-fouling paint, resulting in the practical destruction of the useful qualities of both. The composition of the several coatings of lacquer applied to the external surfaces of the hulls varies somewhat in character, the first coat being almost pure lacquer, mica or kaolin in small quantities being added to the succeeding coats to increase the covering power.

The data relative to the production of this natural varnish, the mode of its preparation, etc., is very meagre.

It is gathered from the *Rhus vermicifera*, or called by the Japanese, *urushi-naki*, which grows to a height of about thirty feet, and at the age of forty years is about forty inches in diameter. It reaches its greatest perfection in eighteen years, producing then the largest yield of lac or varnish. The climate and soil of at least one-half of the area of the United States is as favorable for its growth as that of its natural home in Japan.

The lac is obtained by making a number of horizontal incisions in the bark, in a manner similar to the "boxing" practised for the gathering of the sap of the *terebintus* or turpentine tree, at any time between April and October. The lac obtained later than this is thin and viscid and can be collected only with difficulty.

The lac tapper cuts from six to eight horizontal gashes in the bark upon opposite and all sides of the tree, up and down for a

few feet in height, and when about a dozen trees have been thus tapped the tapper begins to collect the sap which has oozed out from the cuts, which at first is milky white and thick, but through exposure to the air becomes first dark brown and finally black.

This crude lac is called *ki-urushi.* The tree is hacked in this manner for from sixty to eighty days, or until it dies, when it is cut down, the bark and sap-wood removed and steeped in hot water, which extracts the last remnant of the lac, not more than a half pint, which forms the poorest quality of the lac.

The lac is purified by straining it through cotton stuff, grinding it on a paint slab, mixing it with water and then evaporating the latter by heat.

The purest sort is bleached in shallow dishes in the sun. The best quality is called *nashyi-urushi ;* the next poorer kind, *henki-urushi :* and the unbleached, *jeshime-urushi.* The black varnish called *roiro-urushi* is made from the crude lac. There are about twenty different kinds and qualities of the varnish in the market, of which the above are the principal ones.

The operation of varnishing by the Japanese is essentially different than with us, the Japs applying their varnish directly to the wood, less frequently to copper and unglazed stoneware and porcelain. When applied *directly* to tinware the varnish does not adhere, acting in this respect similar to the action of paint upon zinc or galvanized surfaces. This difficulty could no doubt be overcome by first coating the ware with an oxidizing wash that would surface the ware and cause the varnish to adhere firmly. Pure light colors and white cannot be produced from Japan varnish. When applied, the varnishes are generally brilliant black, dark-colored, impure vermilion, or impure dark green or dark gray. The drying is done in a moist atmosphere in twenty-four hours, and if the articles to be dried are placed in closets or close rooms to dry, means must be provided for wetting down the walls and floor, a condition wholly unlike the drying process of paints and varnishes as practised by us.

Various colors are produced by mixing the pigments with the *jeshime-urushi* quality, until the color is formed.

A very fine piece of art lacquer-work requires eighteen coats; it never fades with time, but rather improves in color with age. It bears a high temperature, and is totally unaffected by hot water, sea-water, acidulated mixtures, spirits, etc. The crude lac is corrosive to animal tissues, and care is requisite in handling

it, the sores formed upon the hands being identical to those from nitric acid, and are difficult to heal.

No trustworthy analysis of the crude lac is available. The manipulation of the crude lac to form the various lac varnishes in the Japanese market is a trade secret, and as zealously guarded as any that surrounds the mysteries of our own varnish and paint manufacturers, who practise the boiling of linseed-oil through the bung-hole of the barrel, and who provide us with a variety of paint compounds that are generally misnomers as protective coverings for any materials over which they are spread.

These trade secrets are not impenetrable, whether they be Japanese, English, or other nationality. The secret processes surrounding the manufacture of Russian grained leather were readily brought to light by the Yankee tanner who did duty, for the nonce, as minister plenipotentiary; and that of Russia iron by the practical ironworker in the guise of a gentleman of fortune, when a visit of curiosity to the Russian manufactory was rewarded by just one momentary glimpse of the sandwiching of charcoal dust and iron sheets, and their subsequent mastication under the trip-hammer.

Given the crude material, the secret process, and the need of the hour, the practical and successful result will not be long delayed. This at present appears to be the status of the Japanese natural varnish.

It is hoped that the efforts of our future consular service will be as energetic as in the past in searching out opportunities for the development of American industries, and reporting upon the state of the arts, agriculture, manufactures, and commerce in the several foreign countries to which the consuls are assigned duty, and which have borne fruit in the valuable volumes of Special Consular Reports issued by the Bureau of Statistics, Department of State for the United States. If this practical branch of political science is pursued as in the past, it will not be long before the cultivation of the Japanese varnish tree will be a feature in the treeless wastes of our Western prairies, and a much-needed and valuable addition be made to our practical knowledge of how to protect our iron and steel structures from corrosion and decay, that will be hardly secondary to the efforts of Bessemer, Siemens, Wiley, the Cramps, Fritz, and others in creating them.

In conclusion the whole question of how best to protect iron and steel from corrosion in all the varying conditions that the

wants and usages of to-day demand, seems to resolve itself into a category of Don'ts, as the best method of answering, it to wit:

Don't use anything but common iron.

Don't have any scale on that.

Don't use anything but the best iron and steel.

Don't polish those.

Don't paint it with anything but pure linseed oil and oxide of lead, or graphite, manganese dioxide, or lamp-black paints.

Don't let the air get to it if it is damp.

Don't keep it from the air if the air is pure and dry.

Don't let sea air, sea water, acidulated or sulphurous, ammoniacal or other fumes and liquids have access to it.

Don't think it unnecessary to protect it in any case, because swords, armor and other bright articles of iron and steel have been found uninjured by rust after an exposure of over 500 years with no other protection than that afforded by the closed room in which they were placed.

Don't think your own product would not under the same conditions last as long as the piece of iron that was walled into one of the burial chambers of the Pyramid, 3,000 years ago.

Don't put it into any location where it cannot be inspected and its true condition ascertained at any time, by anybody, your successor in the trust not excepted.

Don't think that magnetic oxide, electroplating, enamelling or any other method of protection, will take the place of constant inspection, even if the coating is fired on by a columbiad.

Don't imagine because Cleopatra's Needle has had to have a coating of wax to protect it, that it is not a good material to apply to other substances than granite.

Don't let the cost and interest accounts be the governing factors in the case of protecting any metal superstructure on whose continuity and strength human life and safety depend. The old story of "for want of a nail, the shoe was cast, the horse disabled, a battle and kingdom lost," finds too many parallels in the engineering practice of to-day, until in some cases we seem to need protection from the engineer quite as much as from the decay of the materials with which he experiments.

Don't imagine that Macaulay's New Zealander, who has sketched the ruins of England's power and greatness, and has come to the New World to see how we have fared from the gnaw-

ing teeth of time, will not recognize amidst the ruins of our Statue of Liberty, Brooklyn Bridge and other monuments of our progress, the ominous streaks and stains due to the corrosion not only of iron and steel, but that of the better class of metals, and will exclaim, "We are wiser in our generation, and fear not, and can control these forces that, like the Arch, slumber not nor sleep."

NOTE.—*The Railroad and Engineering Journal* (Forney's), Vols. 64 and 65, 1890 and 1891, has a series of articles: "Chemistry applied to Railroads," by Dr. C. B. Dudley, chemist, and F. N. Pease, assistant chemist, Penn. R. R., Altoona, Penn. A part of the series relate to paint, its chemical nature, application, adulteration, and other qualities as applied to *wooden structures*, and give the results of a long extended series of experiments on paint as applied to railway purposes, that are well worthy of perusal. The papers are published exclusively by the above journal, beginning September, 1890, Vol. LXIV., paper No. 11, on "Paints." No. 12, October, 1890, on "The Working Qualities of Paints." No. 13, December, 1890, on "The Drying of Paints." No. 14, February, 1891, Vol. LXV., on "The Covering Power of Paints." No. 15, April, 1891, "How to Design a Paint." No. 16, May, and No. 17, June, "Paint Specifications." No. 18, July, and No. 19, August, 1891, "How to Design a Paint."

DISCUSSION.

Mr. J. F. Holloway.—Mr. President, I have no remarks to make on the paper itself. But I have observed, as I dare say most of the members have who are at this meeting, that in Montreal a great many of the roofs are covered with tin and not painted, and it would seem to me that there must be something very peculiar in the tin, or in the atmosphere, that prevents them from rusting, and I trust you will invite any gentleman who is here, whether a member of the Society or not, who lives in Montreal, to give us a reason of why it is that tin lasts so well on roofs in Montreal. It is entirely different from what we are accustomed to seeing down on the seashore or other places, where tin roofs must be protected with a great deal of care.

Mr. Gustavus C. Henning.—They pay a good price for the material, and get tin-plate; they do not get terne-plate, and they do not get plate covered with lead and zinc or anything else. But they nowadays use only steel plate covered with tin, and pay a fair price for it. That is the whole secret. Of course, there is no salt water in the air here. The method of laying, obviating cracks due to temperature strains, has considerable bearing on the lack of corrosion. All seams and joints are laid diagonally on the

roof, and hence expansion and contraction are better provided for than in the ordinary method of horizontal and vertical seams.

Mr. C. W. Nason.—I do not think that the use of unpainted tin is peculiar to Montreal. I have observed it in many of the Western cities. I think it is uncommon to paint it anywhere, except on the seaboard. Inland, where we are far enough away from the sea to avoid getting salt moisture from it, tin lasts fairly well. I think on the New York Central road, excepting in large towns where coal is used sufficiently to cause corrosion, that it is not common to paint tin.

Mr. Wood, in his paper, under the group of "lotions," as he terms them, has apparently omitted to mention a paint which is sold in New York and elsewhere under the name of "P. & B." It is a solution of asphalt or bitumen, suspended apparently, from the unhappy odor which it emits, in bisulphide of carbon, and although the men dislike it at first, they get used to it, and, once applied, I find it more permanent than any of the paints that I know of. I have used it out of doors and also for tank use. I remember an instance of a railing out of doors which had been painted for a number of succeeding years with coal-tar paint, and where the metal had been so neglected that it had rusted through in blotches and spots. Two applications of this paint were put on four years ago, and it is now as good as when first applied. It has been found equally good for tank use. It resists oxidation, and remains intact as well for brine as for water. It is so well known that I think many of the members here must be familiar with it, and I would like to know if their own experience has been equally favorable.

Mr. Gus. C. Henning.—On pages 1011, 1012, 1039, 1047, 1062, 1070, of this paper, the author refers to the corrosion of iron and steel, its probable causes, electric conditions, and other matters which are well worthy of most careful study.

"Prevention is better than cure;" and likewise, in large iron and steel structures it may be asked, Why doctor them, instead of keeping them in a healthful condition? Small pieces may, no doubt, be well protected by a rustless coating, which, at the same time, may be ornamental or artistic, and is not injured by handling. Large and bulky materials always receive rough handling, and hence it might be better to resort to methods which keep them in condition, after erection, if possible, instead of dosing or doctoring them in the shops, and then only skin deep. How this

can be done is a subject of recent investigation, and as it is of general interest, I wish to present the facts given below, taken from *The Journal of the Society of Chemical Industry*, February 28, 1894, London.

William Thomson, F.R.S.E., there states:

"I have lately been experimenting to find the best protective coating for iron and steel work, with special reference to structures which are situated in the neighborhood of the sea and exposed to the spray of salt water. . . .

"I observed that the oxide of iron paint, white lead, and the ordinary paints of commerce had comparatively little protective influence on the iron as compared with red lead; the latter showed no signs of corrosion, whilst all the others had deposited a considerable precipitate of ferric oxide. . . .

"I took two small pieces of iron, attached them by copper wires, the one to a plate of copper *E* was from a piece of new mild steel (unworked). The whole of these samples were pushed through a piece of cardboard, so that all could be immersed in any liquid at the same time. By merely touching the copper wires attached to each with the wires leading to a Thomson's galvanometer, the direction, and, to some extent, the intensity, of the current of electricity produced (if any) could be ascertained by noticing to which side the spot of light was deflected and the rapidity with which the deflection took place. On looking at the annexed table it will be seen that *A*, the piece from the iron rivet, and *B*, the piece from the steel hoop, when placed in sulphuric acid and connected to the galvanometer, the rivet was electro-positive to the steel at first, and thus acted the part of the zinc or more easily corroded material; but within a few seconds afterwards the spot of light swung round in the opposite direction, showing that the iron rivet had become electro-negative to the steel. The piece of iron was then placed in contact with the piece from the new steel rivet, *C*, small size, and the other to a plate of zinc (*Zn*), these being allowed to stand in the saline solution. It was remarkable to observe how rapidly the iron in contact with the copper became corroded, and how completely the iron in contact with the zinc was preserved from corrosion or oxidation, not only on the part in the solution itself, but also on that which was not immersed. After a few weeks an electro-deposit of zinc was formed on the immersed portion of the iron plate in contact with the zinc.

"It suggested itself to me as probable that for the protection of bridges and towers it might be advisable to place a large ball of zinc in wet ground in metallic contact with the iron of the structure by means of wires, which I believe would tend materially to prevent corrosion at comparatively small cost.

"I have lately been considering the effects of riveting iron and steel on the one hand, and different kinds of steel on the other, in the manufacture of iron and steel boilers, with a view to find whether the contact of these different materials would tend to cause corrosion in the one or in the other, by reason of the one acting as electro-negative to the other, and so inducing electric action resulting in oxidation and corrosion of one of the plates.

"I lately came across a peculiar effect produced on the Bowling hoop of a steel boiler which had been attached by iron rivets, the Bowling hoop being deeply corroded at certain points, especially in the line along the outside edge.

"I had cut from the steel Bowling hoop a piece of steel $\frac{5}{16}$ inch square by $1\frac{3}{8}$ inch long, marked *B*, and from one of the iron rivets of Low Moor iron I had a piece of similar size cut and marked *A*.

"A hole was drilled through each near the top, into which a copper wire was fixed; for comparison with these pieces I also obtained similar pieces in size and shape, which I marked *C*; this was a new steel rivet; *D* was a steel rivet closed by machine and held in the dies till nearly cold; and the same effect took place, the iron rivet being electro-positive at first and electro-negative afterwards. Exactly the same thing took place when the iron rivet was placed in contact with the second steel rivet *D*, closed by machine; but when placed in contact with a piece of 'mild steel, new, unworked,' *E*, the iron remained from the first electro-negative, the mild steel being positive.

"The general results, as shown by the table on page 1069, are that the three samples of steel *C*, *D*, and *E*, all react strongly among themselves, and therefore it seems to be quite as likely that one steel plate may act as electro-negative to another steel plate, as it is for a piece of malleable iron to act as electro-negative to a steel plate, although it may be taken as a rule that steel corrodes more readily than iron when placed under unfavorable conditions.

"Copper with all the samples was '+' in caustic potash and '—' in sulphuric acid and chloride of ammonium solutions.

"The + sign corresponds to the electro-positive, i. e., the zinc element which becomes corroded, whilst the sign — corresponds to the copper element, which does not corrode.

"Much interesting work in this direction has been made by Mr. Thomas Andrews, Assoc. M. I. C. E., F. C. S., etc., vol. lxxvii., Trans. I. C. E.; also vol. lxxxii., Part IV., 1883, 1884, and 1885.

Combinations.	Sulphuric Acid. 1 part acid + 9 parts water.	Caustic Potash Solution. Sp. Gr. = 1.311.	Chloride Ammonium Solution. Sp. Gr. = 1.033.
A and *B*.....	+ *A* at first. + *B* afterwards.	+ *B* at first, the current rapidly going down to 0.	+ *A* very strongly on continued action. + *B* very weak, almost nil.
A and *C*.....	+ *A* at first. + *C* afterwards.	+ *C* very strongly.	+ *A* very strongly, afterward the current decreased to 0, but no reversal.
A and *D*.....	+ *A* at first. + *D* afterwards.	+ *D* very strongly.	+ *A* at first weakly, afterwards + *D* weakly.
A and *E*.....	+ *E*.	+ *E* very strongly.	+ *A* strongly at first, gradually becoming very weak, but showed no reversal.
B and *C*.....	+ *C*.	+ *C* at first fairly strongly, then + *B* fairly strongly.	+ *C* strongly at first, afterwards very weakly.
B and *D*.....	+ *B* slightly, + *D* afterwards.	+ *B* at first fairly strongly, then reversing to + *D*, reversing again to + *B* rather strongly.	+ *B* strongly at first, afterwards + *D* weakly.
B and *E*.....	+ *B* momentarily, then + *E* afterwards.	+ *B* momentarily, then + *E* very weakly.	+ *B* strongly at first, afterwards weakly, but no reversal.
C and *D*.....	+ *C* momentarily, then + *D* afterwards.	+ *C* very strongly.	+ *C* strongly at first, afterwards weakly, but no reversal.
C and *E*..	+ *C* strongly at first, afterwards becoming extremely weak.	+ *E* fairly strongly.	+ *E* strongly at first, afterwards weakly, but no reversal.
D and *E*.....	Sometimes on one side and sometimes on the other.	+ *E* very strongly.	+ *D* strongly at first, afterwards weakly, but no reversal.

"He shows that in some combinations of steel of different kinds with each other and with iron, that the one element, after being electro-positive, gradually becomes electro-negative, and this change will take place many times during contest with each other; in other cases the relative galvanic positions of the two pieces of metal remaining constantly the same. Under such conditions it seems an advisable thing that in making boilers some use should be made of this simple means of determining whether the one metal intended for use in the boiler would be likely to act strongly upon another when both are placed in galvanic contact with each other in a saline solution."

It is a well-known fact that when two metals form an electric

couple a current of greater or less force is established, in conjunction with which there is chemical change. This change, as a rule, is what is popularly called corrosion. It has been asserted that all chemical reactions generate electric force, and all that is required to start it is a proper medium to act as a carrier of the electric current.

In the case of iron and steel this medium is a dilute solution of acids in water, and sulphurous acid is a most ready re-agent, but many others act equally well.

As the atmosphere contains more or less acids of various kinds, which are readily absorbed by watery vapor, we can readily see why, when this vapor settles on iron or steel structures as well as others, chemical and electric action should take place at once.

Locomotive engines, steamboats, excreta of animals and vegetable decomposition all give off acids in greater or less quantities, and as these, absorbed by moisture of the air, are precipitated on metal structures, electric action occurs immediately. This action cannot be arrested by any method of protecting the metal by a rustless coating, for with the slightest defect in it moisture will enter by capillary force, and concealed corrosion will grow apace.

The proper remedy to apply is, no doubt, to supply an electro-positive metal, which has much greater electrical affinities than any used in the structure, and then place it in such position and relation to the structure that it will be destroyed and the structure left intact. All that becomes necessary to protect the structure permanently is to renew this mass of metal as it becomes dissipated, and see that its electric connections with the structure remain perfect.

*Mr. M. P. Wood.**—As Mr. Henning aptly remarks, "Prevention is better than cure," but when the doctors fail to cure, or even to properly diagnose the case, who shall decide, and what sort of a chance for life has the patient, be he man, bridge, or steamship?

The tenets of a particular theology are, that Death came into the world by sin, and the writer thinks that possibly oxygen must have come along at the same time, on the general principle that misery loves company; and in providing for the extinction of

* Author's Closure, under the Rules.

spiritual matters, it was wisdom to also have the material elements disposed of.

Mr. Henning's discussion relative to the electrical changes of iron and steel under various circumstances, as detailed in Mr. Thompson's experiments, have been anticipated and referred to in my article, a part of which was overlooked and left out of the paper as presented to the Montreal meeting. Had Mr. Henning been aware of the full text of my article, his discussion would have no doubt been upon a different line; between us both the matter is very fully stated, and need not be further commented on by me.

The use of easily oxidized metals in marine, land, and locomotive boilers to prevent corrosion, was primarily intended to prevent the formation of scale, and its use for the partial suppression of these two evils, if not entirely satisfactory in all cases and with all kinds of water used for steam purposes, is at least generally beneficial, and should be more extensively employed. Zinc has been found the best metal thus far experimented with, and if proper attention is had to frequently blow off or blow out the impurities collected by the decomposition of the zinc, the boilers are benefitted by its use; but if this proceeding is neglected, there is but little difference whether the scale formed is solely a compound of mineral matter and salts in solution or has some metallic matter in its composition.

The decomposition of zinc is always attended with galvanic action, and how far this element may act in preventing the formation of scale or the corrosion of the materials of the boiler, or whether its action is chiefly mechanical, remains to be settled by further and more defined experiment.

It is but a few years ago that an anti-incrustator was extensively experimented with by one of the leading railways of the United States, where a few wires and metallic points and connections made to the dome and other parts of the boiler were expected to gather the frictional electricity generated by a boiler under steam; and, in connection with the galvanic action excited by the use of impure water, threw down the impurities, one and all, ready to be blown off. The use of zinc plates was finally thought to be necessary in the chain of operations, and as this metal was found to be in most cases equally, if *not more, effective without* the mechanical device, the latter soon disappeared as a factor in the case.

That there is a continual electrical action of a most complex character present in all boilers under steam, can scarcely be doubted; and the same action, but less apparent, is possibly present in all constructions of iron when the different members, formed of iron and steel of various compositions, made by different processes, after various torturing methods of manufacture to bring them to the desired shape, are assembled and put into duty under strains and conditions foreign to their nature. It would be strange, indeed, did not some electrical energy manifest itself, and call for some palliative if not protective means of arresting decay. (See page 1037.)

How far an iron or steel roof or bridge truss, viaduct, or other many-membered metallic structure could be satisfactorily wired and connected to a mass of electro-positive metal, the decomposition of which should protect the structure, remains a problem for the coming race of engineers to solve. Thus far we have but little data to leave as an inheritance for their guidance.

The Engineering Magazine, Vol. 7, No. 2, May, 1894, page 297, has an article from Henry Wentz, Ph.D., relative to the preservation of metals subject to the corrosive action of mine waters, that presents a radical change from the practice of the day in regard to this difficulty.

It is well understood, the corrosive action in these cases is an electro-chemical one, due mainly to the presence of copper and other sulphates derived from the oxidation of pyritous minerals as the mines are opened to the air, instead of a simple oxidation of the metal.

In many mines this trouble is so serious as to attack not only the mining pumps, water column, and all the connections, but extends to all of the mine iron-work, ladders, dump-cars, track, tools, etc. Several methods of preventing this action, wholly or in part, are in use, viz.: tarring and asphalting the minor parts of the underground machinery and appliances, and the use of copper, brass, bronze, and other special alloys for the important working parts of the same.

Dr. Wentz proposes as a remedy for this corrosion, whether produced by superficial or internal voltaic currents, the electro-polarization of all the metal-work of a mine as a mass, which shall compose the negative terminal of a much stronger current, generated by a dynamo, while the corrosive liquid acts as the bath for the positive terminal of the much stronger current from the

dynamo, which is carried down into the lowest mine sump and terminates in a body of hard burned coke, being, in fact, a complete exemplification of the modern medical dogma, *similia similibus curantur.*

Could an alloy of some metal with iron or steel be found that would, under atmospheric or other exposure, *be passive* to the attack of oxygen or any corrosive gas or compound, it would be of inestimable value to the world; and if the presence of a small amount of the metal nickel in the plates of an ironclad man-of-war enables it to bid defiance to the heaviest bolts from our modern high-power guns, some future experimenter may yet find the combination of metal that will resist the insidious attack of corrosion that unchecked will be more likely to send the ironclad to the bottom of the sea than any shot fired with hostile intent.

Mr. Nason mentions the satisfactory use of a special brand of paint under the trade-mark of P. & B. I have no experience with this brand of paint, though if it contains, as Mr. Nason thinks, bisulphide of carbon as the vehicle or medium for the pigment, which apparently indicates that natural bitumen or asphaltum is the base or pigment employed, I cannot see why its use should be in any great degree preservative against the corrosion of iron. Sulphur in all its forms is eminently the enemy of ferric substances, as much so, in fact, as oxygen, and, spread over the surface of iron or steel, it could hardly fail, from its close association with the metal, to promote the corrosion that its presence seeks to avoid.

If the pigment with which the bisulphide of carbon was combined was relied upon to correct any corrosive influences on the metal covered by it, or if the bisulphide was used solely as a dryer to the other liquid medium of the paint, I should still think its use would be found unsatisfactory, as the sulphur, if it did not attack the iron or the pigment, would certainly cause whatever oil was combined with it to form the paint to speedily decay, and leave the structure wholly unprotected until the old coating was scraped off and a new coating put upon it.

There are many paint compounds upon the market under all sorts of trade-mark protection or notoriety. Many of these have merit as protective coatings for iron, but the use of them is so comparatively modern that no record of much over ten or fifteen years is to be had; and no paint under such a short period of test

can claim to rank as the Moses that is to lead our perplexed engineering fraternity out of the troubled land where corrosion stalks almost unchecked.

The Detroit Graphite Manufacturing Company are introducing a special brand of graphite paint under the trade-mark of L. S. G. Graphite Paint, prepared from a natural substance (amorphous graphite) mined in the Lake Superior region of Michigan, that appears to possess remarkable powers of resisting heat, acids, water, alkalies, brine, and sea-water, sulphur fumes, etc., all of which applied to surfaces coated with it under test conditions and practice have been resisted with such satisfactory results as to warrant the guarantee that its use is a sure protection against rust or corrosion of any metallic objects under any circumstances and in any climate.

Wooden surfaces coated with it and exposed to heat will char without injury to the paint. A common cotton two-bushel grain-bag was painted two coats with this paint in July, 1892, which has been kept constantly filled with water ever since without leaking. It has been frozen solid twice without injury, not a drop of water has leaked, and it is apparently in as good condition as when first painted, and good for many years more of similar service.

The experiment of coating boiler tubes with graphite paint to prevent corrosion, mentioned elsewhere in this article, was made with this L. S. G. brand of paint, and after a year's trial the tubes are in almost a perfect condition without *corrosion or scale*, while the tubes not painted in the same boiler are badly scaled and pitted from the action of the water. This result is remarkable if for no other reason than the extraordinary preservation of the oil or liquid medium of the paint from the combined agencies of moisture, heat, pressure, and scale qualities of the water used.

Under almost identical conditions flint stones or silica melt down to form water glass. It remains to be tested, how far the use of such paints will increase the evaporative power of a boiler; not alone by preventing the formation of scale, but by the added power of the dark pigment to transmit the heat to the water, being a new application of the well-known law, that black objects absorb and radiate more heat than lighter colored objects whether in daylight or at night, and why the *inside* of a steam boiler should be an exception would be an anomaly.

An analysis of the pigment from which this brand of paint is

made, by Prof. Albert B. Prescott, Professor of Chemistry, University of Michigan, shows that it is entitled to be classed as one of the best of mineral ore pigments, the use of which is proving more satisfactory in all respects than that of the oxides of iron.

	Per Cent.
Carbon graphitic	28.39
Iron soluble as $Fe^2 O^3$	0.41
Iron insoluble as $Fe^2 O^3$	3.81
Aluminium as $Al^2 O^3$	16.90
Calcium as Ca O	0.47
Magnesium as Mg O	0.52
Silica as $Si O_2$	46.97
Moisture	0.13
	97.60
Carbon dioxide, combined water, sodium compounds, loss, and undetermined matter	2.40
	100.00

DXCIX.*

A NEW METHOD OF COMPOUND STEAM DISTRIBUTION.

BY F. M. RITES, PITTSBURG, PA.

(Member of the Society.)

AT the last meeting of the National Electric Light Association in Washington a committee on data read a very interesting report, from which the following extracts are taken:

"It is somewhat unfortunate that it is the duty of this committee to procure and report facts instead of fancies. We listen here to many fine economies and ingenious methods of saving, and if we could report what a given installation ought to do theoretically instead of what it does do practically we could render a report more in keeping with the progressive spirit of these meetings.

"We are requested, however, to furnish this convention with the facts regarding the amount of coal used in actual practice to produce a given amount of electricity. The information was obtained by correspondence and furnished the committee in amperes, volts, and hours on each circuit, and the amount of coal used covering this period, including that used for banking fires, etc. The aggregate electrical output for twenty-four hours was then calculated and compared with the total amount of fuel used, giving the watt-hours per pound of coal. All improbable and apparently erroneous reports were discarded, and the tabulated statement herewith comprises intelligent replies from a large number of the electric stations, including many of the leading corporations.

"The great saving in operating large units is plainly shown in the report; . . . the best results do not compare favorably with the results secured in generating power for manufacturing purposes.

"In order to facilitate this comparison, we have prepared a

* Presented at the Montreal meeting (June, 1891) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

table based on 90% mechanical efficiency in the engine and the same efficiency in the dynamo.

"Then $\frac{(746 \times .90 \times .90)}{(\text{coal per hour})}$ = watt-hours per pound of coal.

"From this estimate of engine and dynamo loss, one and one-half pounds of coal should produce 402.24 watt-hours. We have a report from the Chelsea Jute Mills, of Brooklyn, N. Y., covering a period of six days, where an average of 653.3 indicated horse-power was developed from a coal consumption of 1.482 pounds per indicated horse-power.

"If our percentage of efficiency is correct, and if we could have a fairly uniform electrical out-put, this plant ought to be able to produce over 400 watt-hours, or double that of the most favorable report given us, more than double that of the next best report, thirteen times the efficiency of the plant making the lowest report, and between four and five times the average efficiency of the whole report.

"We hope future reports by this committee will show a much better average than 88.4 watt-hours for one pound of coal."

To the first natural feeling of doubt succeeds the conviction that this report is true, and the question arises, Why should it be so? It is impossible that competent engineering ability should be confined exclusively to the manufacturing industries.

It cannot be assumed that the average intelligence of the designers and operators of electric light stations is inferior to that displayed in establishments of different character, and yet the enormous discrepancy between the actual results and those which should be realized surely deserves some attempt at explanation.

It is but proper to note that the committee has chosen the record of a very high duty as a basis of comparison, and that the nature of the exacting service of electric light and street railway plants precludes the possibility of a close approximation to the highest economy under more favorable conditions; but these figures are entirely unexpected and incidentally somewhat ridiculous considering the energy with which the last per cent. of efficiency of the electric apparatus is insisted on by its users.

Possibly some reason for such a remarkable state of things may be found in the miscellaneous engineering errors which usually follow an ignorantly wasteful policy, but these are as frequently met in other power plants. Perhaps, also, stations

improperly proportioned and generally unfitted for economical competition may be found in the list, but these are far from sufficient to account for such universal failure to realize even a moderate degree of efficiency.

There seems to be but one general explanation applicable to electric light or railway stations which can account with any degree of probability for such extravagant fuel consumption, and that is the excessive wastefulness of the steam engine under varying conditions of load.

With the small clearances of the average Corliss engine there is little left for the single cylinder except the proper variation of the cut-off to suit the load; and with the same grade of high-speed engine, with its comparatively greater clearance, little more can be expected than a simultaneous variation of all the functions of the steam distribution, after the manner generally adopted. For a multiple-expansion engine, however, the customer pays a much higher price, and has the right to expect reasonably good results in fuel economy, which, however, he seldom gets in the electric industries.

It is so generally known as to require no demonstration here, that the cause of the increased efficiency of the multiple-expansion engine lies in the reduction of cylinder condensation through a proper division of the range of temperature between the cylinders. Hence it is evident that the efficiency can be maintained only with a constancy of the division of the temperature. That this is seldom done is shown by the indications of the receiver gauges of the average multiple-cylinder engine in electric power service, where the vacuum frequently passes back through a second receiver.

The indicator card, however, is the complete revelation of the many curious features of abnormal steam distribution which must of necessity lower the efficiency often below that of a simple engine.

Designers have been usually content to provide a mechanism which under approved but seldom available conditions should produce an efficiency to compare favorably with other recorded results, waiving responsibility for failure to maintain that efficiency under varying conditions, and attributing it wholly to the inconstancy or total absence of the requirements.

Recently there has been some interest taken in the subject of uniformity of economy under varying loads, but, with perhaps a

single exception, the mechanical designs placed on the market have consisted of minor modifications of existing types rather than the evolution of distinctive features, based upon and sufficient for the conditions imposed. Published results with tabulated economies are often misleading by restriction of the range of conditions.

Recognizing the increasing necessity for a design of steam distribution which shall overcome the wasteful tendencies which have been described, the following method has been devised by which the proper division of temperature between the cylinders and the efficiency of the engine may be maintained by means of a uniformly unequal variation of the interdependent valve functions, and which has the advantage of adaptability to almost any existing character of mechanism.

Fig. 311 represents a set of compound indicator diagrams under conditions, as to load, which are assumed to be proper for the highest economy.

Brief attention is called to the points of excellence in these cards in order that the valuable features of the peculiarly varying functions may be appreciated as they are described.

The fall of temperature in the two cylinders is substantially inversely proportional to the surfaces exposed for condensation in the respective cylinders, after due allowance has been made for the relative rapidity of the fall and recovery of the temperature. The compound curves, *EH* and *GH*, representing the division of the temperatures between the cylinders, are substantially continuous and show a practically limiting temperature in either cylinder not encroached upon in the other; that is, the lower extreme of temperature in the high-pressure cylinder is practically the higher extreme of temperature in the low-pressure cylinder. All the compression curves, including that into the receiver, approach but do not reach the initial pressures, for in practice it is found unadvisable to correct exactly the loss due to clearance by means of compression, on account of a greater secondary attendant loss through displacement of the division of temperature, except the engine be condensing.

The following method of constructing these cards in detail will serve to indicate the prominent governing features of the diagrams.

The line *AB* represents the higher extreme of pressure and temperature, and the line *CD* represents the lower extreme.

EF is the line of division of temperature determined by the proper laws. With a good, generous high-pressure clearance the compression curve *BG* is drawn to a point not higher than *EF*, and from *E* and *G*, with a knowledge of clearance and receiver volumes, are drawn the curves *EH* and *GH* to their intersection *H*, which is the point of cut-off of the lower pressure cylinder. The low-pressure and high-pressure expansion curves follow as a matter of course, and the low-pressure com-

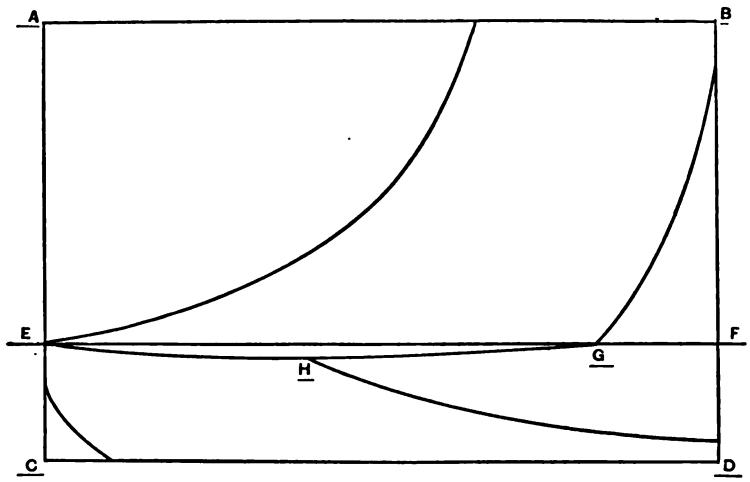

FIG. 311.

pression curve may be located independently of the other features of the cards, but with due regard for the slight influence it may have on the terminal of the high-pressure expansion curve.

Individual designers may vary the method of the primary construction of the cards without interfering with the essential principles under consideration, so long as the results in practice agree and are high in the scale of engine efficiencies, but the real problem to be solved is the proper manipulation of the cards under varying conditions of load and steam pressure, and a variation of the areas of the cards while maintaining the proper location of the division of temperature between the cylinders. In meeting this problem the ordinary plans of governing through variations of the high-pressure cut-off alone, or maintaining a uniform equality of cut-off in both cylinders, is certainly not permissible beyond a very narrow range.

Figs. 312 and 313 represent lighter and heavier loads respect-

ively. With the lighter load the line *EF* falls, and with the heavier load it rises, much faster than if both cut-offs were varied alike, but not nearly so fast as if the high-pressure cut-off alone were varied. In Fig. 312 it will be noticed that the point *G*

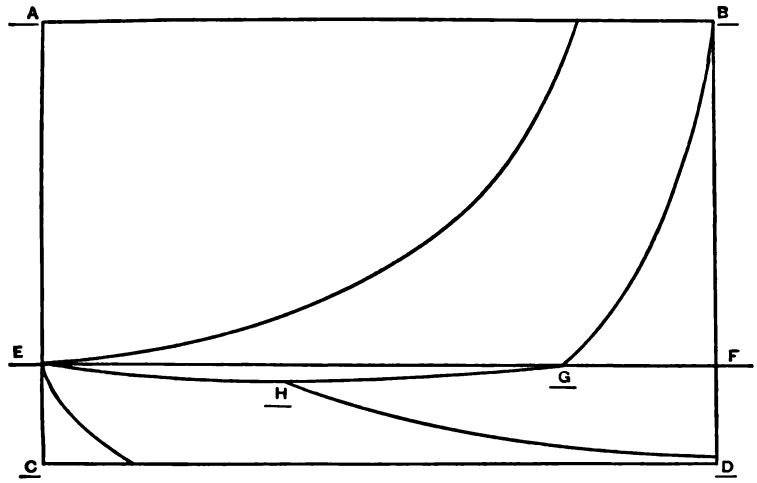

FIG. 312.

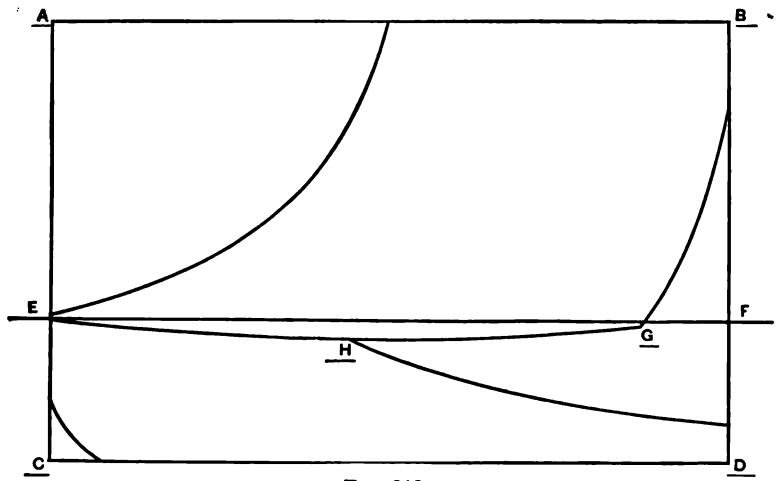

FIG. 313.

approaches the line *EF*, and recedes from it in Fig. 313, while the curve *EH* becomes relatively shorter than *HG* as the load decreases. Both compression curves vary in period and degree, and with extremely light loads either may form a loop at its upper extremity. The chief interest centres in the movement of

the point G, and, theoretically, it should vary along the compression curve GB (see fragmentary card, Fig. 314). Instead, however, of adhering rigidly to this path, the point should follow a slightly different course, as shown in Fig. 315, with the end in

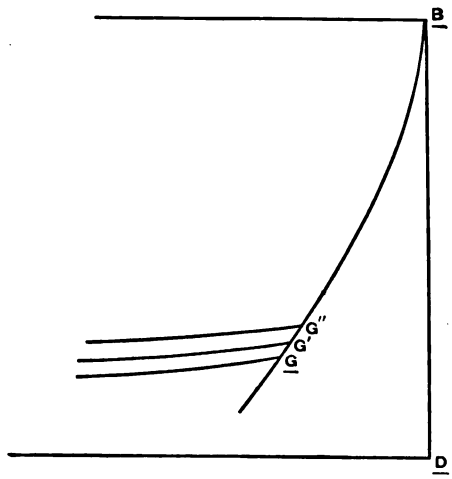

FIG. 314.

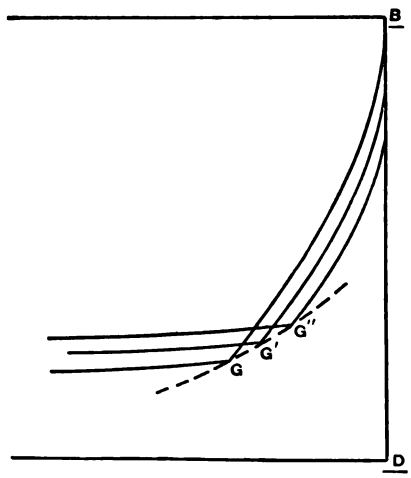

FIG. 315.

view of assisting the proper location of EF under the varying conditions of load.

In Fig. 316, all the cards represented in Figs. 311, 312, and 313 are superimposed, and will serve more readily to compare

the peculiar features of the similar functions of the valve mechanism.

It will appear, then, that the method consists in a simultaneous variation of all the valve functions in the same direction but with different degree and period, and with particular reference to the position of the line *EF*. Also that although an exact compression to initial pressure is usually unavailable, except at a single position of governor adjustment, yet the greatest departure from such a degree which is resultant on this method is immaterial and is advisable for its beneficial reaction on the position of *EF*.

With a governor controlling all the valve functions, the range

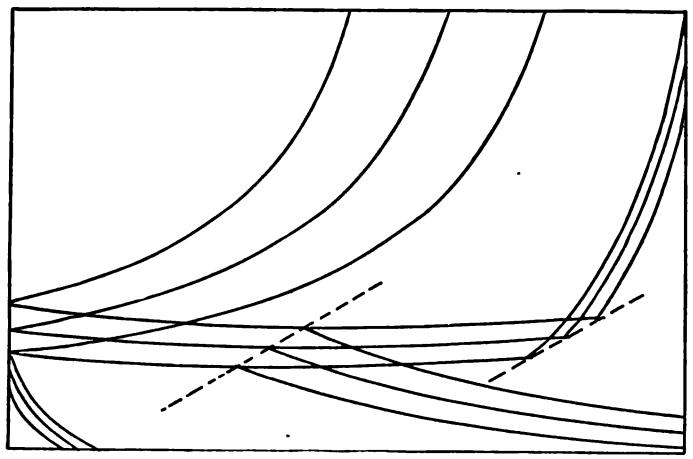

FIG. 316.

of variation of high-pressure cut-off is reduced to a minimum, for the areas of the cards are varied in all directions at once, and, incidentally, the danger of an engine running away with a light load is avoided.

The above would be incomplete with nothing beyond a mere allusion to the low-pressure compression curve. Convenience would suggest that it be constant in character, but advantage is taken of this opportunity to insist that its point of origin should likewise vary with the other functions of the valves. Such an action serves a double purpose—to assist in the maintenance of the position of the line *EF* while reducing radiation to the condenser or atmosphere with the lighter loads.

If, however, this correction be overdone, as is often the case where a single connection to a shaft governor controls all the valve functions of both cylinders, an excessive loop in the compression curve is formed. On the other hand if any one of the three compression curves (and these include compression into the receiver) are deficient with the lighter loads, a loop with the low-pressure expansion curve is sure to result.

The designer is thus placed between two fires in this respect; but an error through a loop in the compression curve is of far less consequence than in the expansion curve, and it is by attention to the compression curves alone that an expansion curve loop with its attendant evils can be avoided.

Such a distribution as described may be so designed as to avoid entirely an expansion loop, and it is difficult to understand how the method can be improved; for in each case of readjustment the division of temperature is maintained as perfectly as if the engine, while retaining its dimensions, had been purposely designed to meet the momentary conditions Moreover, the principles here involved, instead of being limited to one or two special types, may be readily adapted to any existing character of engine mechanism.

The foregoing is not claimed as an infallible cure for all of the faults of electric light and power stations, but it is firmly believed that reasonable attention by designers of steam engines to the question of maintaining high economy under variations of load will go far toward enabling the committee on data to "make more favorable future reports."

DISCUSSION.

Mr. Geo. I. Rockwood.—I should like to ask Mr. Rites to explain briefly exactly how he aims to preserve this constancy of the division of the temperature in the two cylinders by his method.

I ask the question because I believe the proposition of Mr. Rites to be absolutely true, if I have a true grasp of its meaning. He states that his aim is to vary the normal range in temperature in each cylinder as little as possible under wide variations in load. But I do not understand the meaning of that paragraph on page 1077, especially the first part, where it says that the fall of temperature in the two cylinders

is substantially inversely proportional to the surfaces exposed for condensation in the respective cylinders, after due allowance has been made for the relative rapidity of the fall and recovery of the temperature. What fall in temperature is here meant? That of the steam or that of the cylinder walls? I should suppose that the temperature of the walls was dependent upon that of the contained steam, the temperature of which varies with the pressure and is independent of the amount of surface exposed.

But, leaving this, it is not clear to me that the new method whereby he proposes to maintain his approximate constancy of the division of temperature with change of load is either novel where true, or true where novel.

Quoting from the paper, page 1077, he says the "method has been devised by which the *proper* division of temperature between the cylinders and the efficiency of the engine may be maintained by means of a uniformly unequal variation of the interdependent valve functions."

To deal with the first part of this remark, there is little in the paper to show what may be a "proper" division of temperature, although the author makes many references to it. Any one who can tell what is a proper range in temperature for each cylinder of a compound engine will confer a great benefit. At present, I think I am safe in saying it is not known; yet, as the best design of compound engine depends in great part on properly fixing this temperature division, there is always something left to chance in compound engine designs.

Turning our attention to the second part of the quoted remark, "the efficiency of the engine may be maintained by means of a uniformly unequal variation of the interdependent valve functions," all will doubtless agree that it is somewhat confusing. After some study I conclude its meaning to be simply that the change in cut-off in one cylinder must not occur as rapidly as that in the other; that the compression, too, in each cylinder must be of unequal amounts. But just how variable the relation of the number of expansions and the number of compressions in each cylinder must be to each other is nowhere stated in the paper. The only information given is that there is great danger of overdoing or of underdoing the matter. We must, therefore, turn to the diagrams shown in Fig. 316 to fully understand the novelty of the system. Now, whether these diagrams are correctly drawn or not is of no consequence vital to the interests of the paper, except in

one very important particular. But in this very particular the diagrams of Fig. 316 appear to me to be absolutely wrong.

It should need no demonstration to prove that the point of cut-off in the low-pressure cylinder cannot be varied in the manner shown in the figure, and the diagrams retain in an actual engine the shapes given them here. In any compound engine a sharp toe will appear on the high-pressure diagram when the point of cut-off in the low-pressure cylinder has been so adjusted as to take from the receiver the same volume of steam put into it by the high-pressure cylinder. If the card having the toe A corresponds with the card from low cylinder having cut-off at A^1, then, neglecting the effects of compression, A^1 is the only point of cut-off which will constantly give a sharp toe as at A, B, and C. Now, as to the modifying influence of a variable compression, not only is the compression in high-pressure cylinder altered, as at C, B^2, and A^2, but it is also altered in the low-pressure cylinder at A^3, B^3, C^3, and the *effect* of the change of compression in one cylinder on altering the point A so as to give a cut-off in low cylinder which will maintain the sharp toe A, B, or C will mostly counterbalance that in the other. But even if it did not, a mere tinkering with the compression will have little gross effect on the size of the low-pressure diagram.

I believe Mr. Rites has stated in a previous paper that "drop" at the terminal of the high-pressure expansion curve represents a considerable loss of potential energy, and that, as the receiver volume increases, "drop" must also increase—that all receiver space means clearance space and implies drop. So I infer from that and from this present paper that he really is following out that line of thought, and is endeavoring to maintain the sharp toe of the high-pressure card, paying much more attention to accomplishing this than to maintaining the constancy of the temperature division. I therefore have referred to the fact that the cut-off in the low-pressure cylinder cannot be made earlier, as at B^1 or C^1, and yet retain the sharp toe at B or C, without producing either a loop at C or a drop at A; and to the fact that "drop" is not necessarily produced by receiver space. But Mr. Rites apparently lays stress on the ability to shorten cut-off in the low-pressure cylinder without producing either a loop or "drop" as forming, so far as I can judge, the chief feature of his new method. Stress, too, is laid upon the importance of getting just the right compression in the receiver, although the curvature of

the receiver compression curve, as drawn in diagrams Fig. 316, is so slight as to be hardly noticeable.

I cannot understand why, from the part played by the receiver compression curve in the new method, it should be deemed of importance.

For all that I can see, Mr. Rites's "New Method of Compound Steam Distribution" resolves itself into a method for maintaining the toe *A* which will not maintain it, and a method for varying the area of the high-pressure card by changing the position of the high-pressure compression curve, which is not novel. So far, then, from having evolved distinctive features, based upon and sufficient for the conditions imposed, it seems rather as if here, indeed, was but a minor modification of existing types.

*Mr. Rites.**—It would appear from Mr. Rockwood's remarks that he misunderstands the drift of the paper, and it will hardly do to take the space to demonstrate all the points to which he refers, particularly since they are mostly foreign to the subject. However, the following notes may assist in clearing up the greater part of these doubts.

(1) Regarding the objection to the term "proper division of temperature," let it be remarked that the writer did not presume to positively state just what this division should be, and probably knew as little about the point as his critic. On the contrary, the article states plainly that the proper division of temperature is assumed, and further suggests that other designers may easily disagree in regard to the fundamental diagrams.

(2) The writer has as yet no data or cards of actual performance of this distribution, otherwise they would have been incorporated with the paper.

The figures are but sketches, and if the critic had investigated further he would have found that the different portions of the same card do not mate; but they serve, as they were intended, to illustrate the idea. On account of the small range of the low-pressure cut-off, and the following high-pressure compression, the variation of the point of the toe of the high-pressure card will be so slight that it could not be noted in the total range of cut-off which is illustrated. It is also maintained, in spite of the assertion that the low-pressure cut-off cannot be varied without altering the character of the toe, that the low-pressure cut-off *can* be

* Author's Closure, under the Rules.

changed *without* altering the toe if the high-pressure compression is correspondingly varied.

(3) Regarding the statement that the variation of the low-pressure compression cannot materially alter the area of the card, it is evident that all will agree that it is perfectly possible to go to an extreme in the matter and so far change the area of the card by such means that the engine may be governed entirely by shifting the point of compression.

(4) The idea that the method is "but a minor modification of existing types" is correct if modified. The method is really a modification of the receiver compound, and it is not claimed anywhere in the paper that there is anything even approaching a new type of distribution.

(5) The conclusion that the method resolves itself into a method of maintaining the toe is also not warranted.

It is not desired to maintain this toe, and nowhere in the article occurs a hint of such an aim; but there is a description of quite a different thing, which the new method will certainly accomplish.

DC.*

CORROSION OF STEAM DRUMS.

BY JAMES McBRIDE, BROOKLYN, N. Y.

(Member of the Society.)

AT the Richmond meeting of the Society, November, 1890, there was discussed this topical question:

Is there any reason why corrosion should be more active in one place rather than in another, inside a steam drum, properly piped, to connect several boilers in a battery? †

The subject was suggested by some experiences of the writer, and provoked quite a lively discussion. Specimens of the cor-

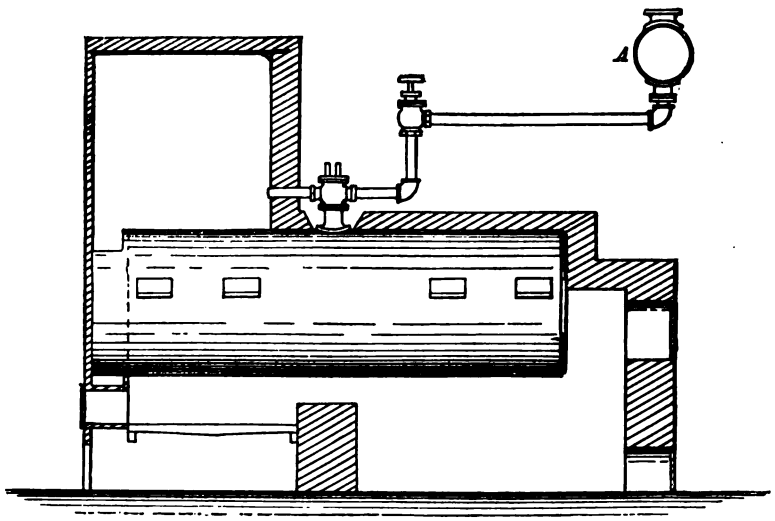

FIG. 317.

roded drums were shown to the members present and examined, and a number of theories were advanced to account for this curious action upon the iron, but no apparently satisfactory cause was

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Vol. XV. of the *Transactions*.

† *Transactions of the American Society of Mechanical Engineers*, Vol. XII., p. 518, No. 436.

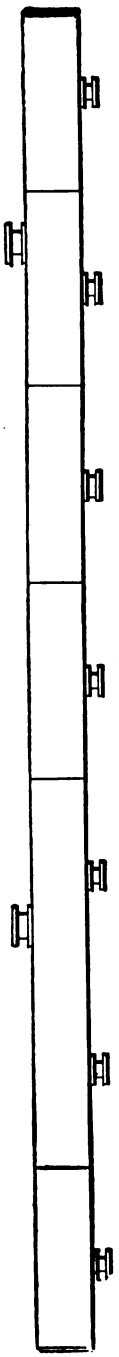

FIG. 318.

assigned. The subject excited such general interest at the time, and since, that the writer determined, if possible, to trace the trouble to its true source, and immediately began such experiments and collected such data as would, in his opinion, lead to this result.

In order that members who were not present at the Richmond meeting may have a knowledge of the case the following description of the drums and their attachments will suffice:

The drums were made of iron plates $\frac{1}{4}$ inch thick and single riveted, were 24 inches in diameter and 48 feet long, and were located and piped to the boilers, as shown in the accompanying sketches (Figs. 317 and 318), seven boilers to each drum. They were erected and put to work in March, 1879, and were kept in continuous service, day and night, six days in the week, until the summer of 1890, when, on account of their bad condition, it was found necessary to replace them with new ones, which were made 3 feet in diameter, and of the same length (viz., 48 feet).

During the first few years that the old drums were in use the boilers were not worked very hard; but as business increased, and more steam was required, the boilers were called on to produce more, until they were apparently greatly overworked.

Previous to the increase of work, however, I made a large number of calorimeter determinations of the quality of the steam, and the mean for some forty or more tests showed moisture equal to $4\frac{5}{8}$%. The demand for more steam continued to increase, and to make matters worse it was decided, in February, 1890, to shut down three boilers at a time for cleaning, stopping one each night and starting one each morning, thus making a continuous cleaning process, so as to avoid the loss of one day each month for cleaning them all at once, as had been the rule. This compelled eighteen boilers to do the work which twenty-one could not do properly.

Early in 1891 it was decided to increase the boiler capacity of the works, which was done by the addition of two vertical boilers of about 130 H.P. each. The two new boilers were put to work on July 1, and were worked in connection with the others, the steam from all leading into one general steam-pipe.

No calorimeter tests were made from early in 1891 until August, 1891, after starting the new boilers, so that nothing definite is known as to the quality of the steam during the time that eighteen boilers were made to do the work of twenty-one. The amount of evaporation, however, based upon the consumption of feed-water, which was metered during a test of 141 hours, and which amounted to about 582 cubic feet per hour for the twenty-one boilers, thus giving about 70 H.P. each, does not seem excessive for boilers of this size, 5 feet by 15 feet (824 square feet). The new boilers furnish superheated steam at all times, and, after this steam mingled with the steam from the other boilers, calorimeter tests showed at times as much as 20% of priming, although sometimes, when the demand for steam was small, the tests showed a slight superheat. As a rule, however, the steam was very wet, the mean for six tests, taken at intervals from August 7 to August 28, 1891, being 6.82% of moisture, and for eight tests, taken from August 29 until September 3, being 10.44% moisture, the highest at any time being 20% and the lowest 1.84%; so it is a fair assumption that with 260 H.P. of dry steam from the two vertical boilers, mingled with the wet steam from the eighteen horizontal boilers, there was such a large percentage of priming that it must have been very much more when only eighteen boilers were in service. The particular stress laid upon these calorimeter tests may seem foreign to the matter in hand, but I am firmly convinced that they are the stepping-stones to the solution of this problem, and have been thus particular in mentioning them. Having a theory, at the time the new drums were put in, as to the cause of the corrosion and the cure for it, I had one of the three new drums piped differently from the others.

It will be noticed from the sketches of the old drums (Figs. 317 and 318) that the 4-inch steam-pipes from the boilers led into the bottom of the drums, no provision being made for the return of the condensed steam and entrained water into the boiler except through the steam-pipes. Two of the new drums were piped similarly to the old ones, and one had the steam-pipes

led into the side (Fig. 323), and a drip-pipe from the bottom of the drum led into the boilers below the water-line, thus keeping the drum at all times free from water.

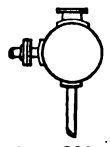

FIG. 323.

All the new drums were kept at work from the summer of 1890 until the 30th of May, 1891, when I had them opened up and made a thorough personal inspection of them. The result of this examination seemed to confirm the opinion which I had formed as to the trouble. The two new drums which were piped similarly to the old ones showed unmistakable evidence of corrosion in the same locations and in the same manner as their predecessors, while the one piped differently did not show the slightest indications of corrosion in any part.

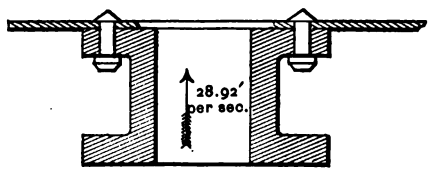

FIG. 319.

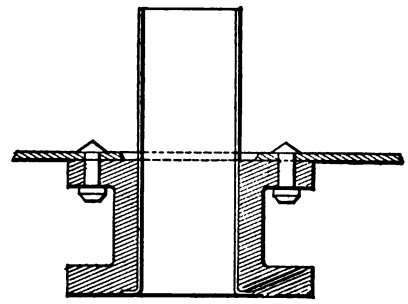

FIG. 320.

On August 2, 1891, following, two months later, I examined them again. The corrosion in two of them seemed as active as

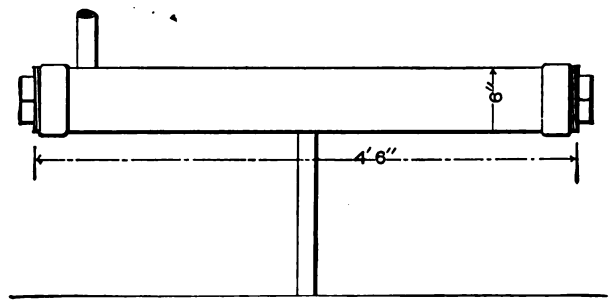

FIG. 321.

ever, while the other showed no signs. I now determined to fix one of those which was being corroded so that it could be kept free from water, similar to the one which was piped on the

side. I therefore had 4-inch copper sleeves made and inserted in the nozzles where the pipes entered on the bottom, and extending up into the drum about 6 inches, as will be seen in the sketch (Fig. 320). I then put a drip-pipe into each end of the drum and led it back below the water-line of the boiler. This is similar to a method adopted by Mr. Louis G. Engel, of the Brooklyn Sugar Refining Company, and a member of this Society, who has had some trouble of a like nature with the cast-iron fittings on the pipes leading from his steam-drums. These sleeves were put in on August 9, 1891, and the drum examined on September 5 following. The appearance of the surface at this time indicated that corrosion was ceasing. The corroded parts seemed to be covered with rust, and there was no place where corrosion appeared to be active.

On November 5, 1891, the factory being shut down for repairs, another examination was carefully made, the result of which confirmed that of August 9. The drum having the sleeves showed that corrosion was ceasing, while the other one, piped like it, with the exception of the sleeves, showed that corrosion was still active, but perceptibly less than before the additional boilers were put to work.

Examinations were again made on December 26, 1891, and July 4, 1892. At this latter time corrosion in the drum with the sleeves had entirely ceased. The drum without the sleeves, piped on the bottom like the old ones, showed that active corrosion was

FIG. 324.

have primed very badly, the evidence of this being that there was quite an accumulation of mud near one end, which was lower than the other, and on which the drip-pipe had evidently been closed. A sample of this mud, dried, is shown, and is no doubt the surface scum and sediment which was carried over when the boilers primed. Being firmly convinced that wet steam or priming, or that the water brought out of the boilers into the drum, together with the water of condensation from the surface of the drum, being thrown up and kept in motion in the drum, was the cause of the trouble, I decided to make an experimental drum, to demonstrate, if possible, if this was true. This drum consisted of a new piece of 6-inch iron pipe, 5 feet 6 inches long, and was located on the top of one of the large drums. (See Fig. 321). The ends were plugged, and the steam was admitted into the bottom at the centre through a 1½-inch pipe, and taken away from the top, at 6 inches from one end, to a feed-water pump, and was used continuously night and day for about nine months. This experimental drum was not covered and had good exposure for surface condensation. There was no drip-pipe, and the water had to get out with the steam on its way to the pump. When taken down I had a cross-section cut from near the centre and then planed in two longitudinally, and which I submit for examination. Photographs are also shown in Figs. 324 and 325, with this paper. It will be noticed that on the inside, above where the steam entered this experimental drum, the iron is corroded and eaten in holes, and has every appearance of the corrosive action seen on the large drums.

The water used in the boiler was "Ridgewood," an exceptionally pure water for boilers, and had mixed with it a large percentage of water of condensation from the copper coils of the vacuum pans. The water was analyzed and nothing injurious to iron was found in it. The fact that the boilers primed so badly was due, probably, to the fact that the water was carried very high in them (as will be seen from the sketch, Fig. 322), and the pressure was very fluctuating, ranging from 40 to 80 lbs. at short intervals. The demand for steam was also very irregular, large demands being made on the boilers at times when the pressure was low; and those who have had experience with vacuum pans know that their capacity for steam is like that of a young pigeon for food, they never get enough. A pan with 300 square feet of coil surface will make away with 345 H.P., and then call for more.

These investigations have convinced me that the method of piping boilers with the end of the steam-pipe lowest at the end next the boiler, so as to allow the water of condensation to run back to the boiler, is a fallacy, and in my opinion not a drop of water, while the boilers are steaming, ever gets back to the boiler by this route, even if the pipes were vertical. The velocity of the steam in steam-pipes of ordinary size is sufficient to carry along with it any water which may be on the surface of the pipes, so that in the case of these drums piped on the bottom, all the water in them had to go out with the steam, and had to go out as water too, for it could not be re-evaporated. Estimating an evaporation of only 2,000 lbs. of water per hour for one of these boilers, I figure

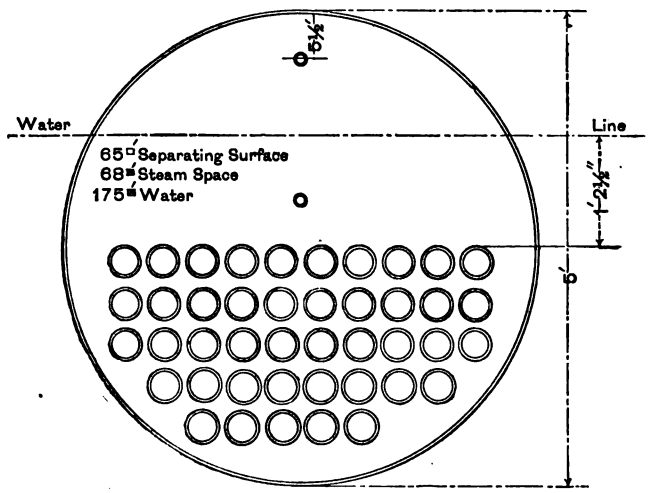

FIG. 322.

the velocity of the steam through a 4-inch opening equal to about 28.92 feet per second. No water could ever pass down through this opening to the boiler, as it would have to start practically from a state of rest at the edge of the opening (see sketch, Fig. 320), and the difference in its specific gravity from that of the steam would not be sufficient to overcome the velocity of the latter. I believe that all drums receiving steam in the bottom should be furnished with sleeves extending up some distance into the drum, and the water taken away with drip-pipes, and that all steam-pipes should be drained away from the boilers, and the water trapped out before reaching the engine.

The conclusions which I draw from these observations and

experiments, which have extended over a period of nearly four years, and have been carefully made, are:

First. That the corrosion is due primarily to excessive moisture in the steam in the drum.

Second. That moisture in this case was due to the priming of the boilers and surface condensation in the drum.

Third. That the trouble was aggravated because no provision was made to take away the water from the drums by drip-pipes.

What peculiar action takes place to corrode the iron, under the above conditions, I am unable to say, and whether it is mechanical, chemical, electrical, or a combination of any or all of those actions, is open to discussion.

DISCUSSION.

Prof. F. R. Hutton.—The accompanying sketch, Fig. 326, is from

FIG. 326.

a 3-inch cast-iron elbow, which stood at the bottom of the descending column of a Bulkley condenser used near Port Henry, on Lake Champlain, N. Y. The bottom of the elbow is worn through to a film, with a hole in it, and the structure of the iron which remains is opened into a sort of columnar form. It is an

illustration of mechanical erosion by flowing water, as the pipe should have been full of water all the time except, perhaps, just at starting. Mechanical corrosion is a very common experience where temperature changes, or changes of shape, can cause the protecting scale of the sesquioxide of iron to be broken away so as to leave a fresh surface to rust, and be again removed.

Mr. John J. Flather.—I would like to ask Mr. McBride if it was not possible for there to have been gritty matter in this water. Water may be chemically pure, or practically so, and yet contain a considerable amount of sediment. I would like to know if in this experimental drum there were not particles of sand or sediment of some nature that might erode the plates somewhat in the manner of a sand-blast; the appearance of the plate before us would seem to indicate that the action was partly mechanical as well as chemical.

Mr. J. F. Holloway.—I would like to say a few words in regard to this. There seems to be a general impression among the writers that have responded to the invitation to discuss the subject, as well as with the author himself, that this trouble has arisen from the erosive action of water, or of the water and such gritty material as it may contain. A careful examination of this plate will show that it is hardly possible that this could be so. Erosive action, such as is shown in the elbow of a pipe, is a gradual and uniform diminution of the thickness of the plate. But if you look at this particular plate you will find that it is honeycombed, or, rather, that the erosion has taken place in spots and streaks, and that it looks like the teeth of a comb, having ridges of iron in between; and that the erosion has not been uniform and smooth. Inasmuch as the making of iron boiler-plate is, I dare say, one of the lost arts, and there are perhaps no gentlemen here who are now making boiler-plates of iron, and this being an iron boiler-plate, I would say that the appearance of the defect as shown would indicate to me that it was caused by cinder in the plate, and that the action of the water, together with whatever gritty material might be in it, had cut out the cinder, and had left the iron. There may be other samples which show differently. But if this was simply a thinning down of the plate by erosive action of water, the same as if rubbed down with sand-paper, it would show a smooth depression, and not a ragged and toothed one, as is shown.

Mr. George R. Babbitt.—My experience is confirmatory of that

of Mr. McBride, as detailed in this interesting paper. While engaged in the manufacture of Corliss engines I had an experience which illustrated the erosion of cast iron by a current of steam and water.

To make them lighter to handle, the exhaust-valves of the larger sizes of engines were cored, thus forming a chamber in the valve. The openings through which the cores were removed were afterwards plugged steam-tight to economize clearance. To finish the valve, it was pinned to the stem and then turned to the diameter of the bored part. In some cases, through carelessness of the workman or bad coring, the pins would penetrate the cored chamber of the valve. In every such case, unless the holes were afterwards plugged, in a short time the bridge or metal between the openings in the exhaust-valve chamber would have the appearance of having been worn as if by a sand-blast, exactly opposite to the location of the open pin-holes in the valve. As no other damage than the added clearance was anticipated by the pin-holes penetrating the cored chamber, it was a mystery what had caused the abrasion of the iron; but upon investigation of several engines it was evident that the escaping steam and water through the pin-holes when the engine exhausted was the cause of all the trouble.

Mr. Carleton W. Nason.—I do not think it is necessary to look for any other than erosive action in this case. The water was Ridgewood. It was nearly free from any agents that commonly injure boilers, and it is hence improbable that any chemical cause should be sought.

Mr. McBride admits in his paper that the boilers were entirely insufficient for the work that they were doing, and that they were being driven considerably beyond the point for which they were designed; therefore it is quite reasonable to assume that they were priming to a greater or less extent, and that there was also carried over with the water a considerable amount of boiler scale and deposit.

It is, however, further stated that the steam pressure at times falls as low as 40 lbs., under which latter condition, providing that the same weight of steam per second was delivered, the velocity under Marriotte's law would be increased to $\frac{80 \times 15}{40 \times 15} \times 29 = 50$ feet per second, approximately.

There being no return pipe for conveying the water which lay

in the bottom of the drum back into the boiler, it is evident that this water, with its dirt in suspension, lay as a stratum, and, necessarily covering the steam inlet pipe, it is quite apparent that steam, on entering at the high velocity shown, came in contact with the water and projected it and the material in suspension against the opposite side of the cylinder. This naturally produced an erosive action quite analogous in its effect to that of the sand-blast, and rapidly produced the effect which we have seen on the plate exhibited by Mr. McBride.

Following the law of what may be termed erosive attack, any small concave surface which might be formed under the cutting action of the water and material in suspension would be rapidly enlarged, and such holes as are shown in the plates would be consequently formed.

With these facts in view, I do not think it necessary to look for anything more than a purely mechanical cause as having produced the results shown.

Mr. McBride.—In answer to Mr. Flather's question about sediment in the experimental drum I will say that this experimental drum was placed immediately on top of the large drum, probably three feet from it, and all the steam that passed into this drum came through the lower one. I never examined it, but when it was taken down it was perfectly clean. By referring to page 1093 of the paper you will find I mention that there was evidence of a large amount of entrained water and condensation in the drum that was piped on the side, principally entrainment, I think. The drip-valve from one end of this drum had been closed for some time, and that end of the drum being two inches or so lower than the other, water collected there, and I found on examining it this deposit of fine clay.

As to the use and action of kerosene in boilers, in my case it resulted in a bill of expense for renewing the tubes. It made the tubes leak on the ends. I put it in to remove the scale, but had to discontinue it because it made the tubes leak.

Mr. J. F. Holloway.—There can be no question, I think, in the minds, certainly, of those who have spoken—I do not know what is in the minds of those who have not spoken—that this erosion here is due to the action of the water. But the question I raised is, how does it come to make such a hole as there is in that plate, unless there is a cinder in that plate?

Mr. McBride.—I found the worst corrosion above where the

steam entered the drum. The corrosion was worse there than at any other place. But I found patches and spots elsewhere. The sample of plate shown was cut out of the part of the drum half-way between the openings where the steam entered, and not directly in the path of any violent currents from the pipes. Some of the other pieces were taken out near the end and some were cut out near the top, so that all along, on the top and on the sides, and where the cast-iron nozzles were riveted on the bottom, there was the corrosion. Some rivet-heads were almost entirely gone. In many of them there was not a sixth part of the original rivet-head left.

Mr. John T. Hawkins.—As the dropping of water will wear away a stone, there ought to be little doubt that the same substance, heated, and caused to continuously impinge at a high velocity upon metallic surfaces, should be competent to wear them away. And as the rushing torrent wears the angular bowlder into a rounded pebble, there is as little doubt that currents of water passing through metallic pipes must wear away any parts of such pipes as are called upon to change the direction of the currents.

Professor Hutton's elbow specimen is a fair example of the latter action, and Mr. McBride's experiments show conclusively enough that the effects produced in his drums resulted from the more or less violent impact of steam, water, or water-laden steam against the surfaces which showed impairment.

In the drums, as at first piped, shown in Figs. 317 and 318, it is quite certain that a part of the bottom circumference, under the amount of priming his boilers generally showed, was always more or less covered with water; and it is not improbable that, generally, under his working conditions, the amount of water lying in the drum was such that the upward currents of steam through the entrance nozzles must continuously throw a fringe of this water violently against the upper surface.

It matters little whether we regard the resultant action as a wearing away of the oxide coating previously formed, to be again formed when the drum was out of use, and thus repeated at intervals of a long service (as advanced by me in a discussion of this subject at the Richmond meeting), or whether such oxide formation is more easily removed from such surfaces by such action than a correspondingly slight portion of the metal itself would be. That the parts of such a drum would be worn away under the

given conditions appears to be certain, and the amount of this impairment of the metal in a given time probably depends upon the proportion of water entrained in the currents, the velocity with which it strikes, and, possibly, somewhat upon the temperature and purity, or freedom from gritty substances, of the impinging current.

If this view of the matter is correct, Mr. McBride's change in the position of the entrance nozzles, from the bottom to the side of the drum, as well as the introduction of the copper sleeves, should, of course, ameliorate the trouble, if not entirely cure it. In either case it amounted to turning the drum into a sort of dry pipe, since both methods provided against the further entrainment in the entering stream of water previously deposited in the bottom of the drum.

The photographs show unequal erosion, as a sort of pitting; but this is, of course, accounted for by the lack of homogeneity in the iron of which the plates were made.

I would suggest to Mr. McBride, in way of further experiment, that he attach both the entrance and exit nozzles to his drum at the top, and arrange in the draining of the drum to leave water in it at all times sufficient to cover, say, the lower quarter of its circumference. Under such an arrangement the steam, or mixture of steam and water, entering at the top and projected downward, would encounter the water in the bottom of the drum, which would probably serve as a shield against wearing away at the bottom, opposite the entrance nozzles; at all events, it would not be amiss to institute this in further experiments if he still has any doubts as to the real cause of the trouble.

Mr. L. S. Randolph.—This paper has been of exceeding interest to me, as it would indicate that water very slightly charged with sediment can produce corrosion by removing the iron-rust as soon as formed.

What observations I have been able to make lead me to believe that all cases of corrosion can be attributed to either chemical action, due to the presence of acid in the water, or to mechanical action, caused, first, by unequal straining, and the consequent bending of the metal breaking off the iron oxide as formed, or, second, by the attrition of particles of solid matter or sediment knocking the iron-rust off, and thus exposing the fresh metal.

Many cases of this mechanical action have been observed by the writer. In this case the presence of mud in large quantities

in the steam drum would readily account for action described; but in the experimental drum, if the writer's understanding of Mr. McBride is correct, the water was obtained principally by condensation, and consequently should have contained very little sediment. The same may be said, as regards sediment carried, for the case mentioned by Professor Hutton.

I have tried several times to determine whether water without sediment in suspension could produce this action, but so far to the contrary, as every case has shown decided evidence of the presence of sediment in suspension in the water, and sometimes as if the action were almost that of a sand-blast, or similar to the cutting out of the elbows of sand-pumps used in putting down caissons for bridge-pier foundations.

Mr. Paul H. Grimm.—Observations which I have made since the Richmond meeting have only confirmed the position which I then took, that is, that the corrosion is due to the mechanical action of the entrained water in the drums; such water carries with it a great deal of gritty matter, which, being projected against the iron, will show the same effect as if corrosion had taken place from chemical action.

We have a great deal of trouble in steam-heating coils, where circulation is intermittent, and very often find cast-iron fittings, as well as the ends of pipes where they are screwed into the fittings, reduced to the thickness of paper. I fully agree as to the remedy of this trouble in steam drums, and I have overcome the trouble in steam-coils very successfully by establishing perfect drainage.

Mr. E. F. C. Davis.—I have read the paper on "Corrosion of Steam Drums" with much interest, but the author has analyzed the subject so thoroughly, and his conclusions are so obvious, that I could not find anything further to say that would add to the interest of the discussion.

I have realized, in my experience around the coal mines in Pennsylvania, a great deal of trouble from corrosion of steam-pipes, which we always supposed was due to the acid which is contained in larger or smaller quantities in all the feed-water obtainable at the mines. From the author's experience it would appear, however, that even the purest water will corrode the pipes and steam drums when carried through in connection with the steam.

Mr. Chas. E. Emery.—Corrosion in steam drums of the kind described in the paper is new to me, but all my experience has

been either with steam chimneys acting as superheaters, horizontal drums, through the lower part of which there is a circulation of water, or vertical drums carefully drained. Some will recollect that the boilers of the Steam Company, of which there are to be eventually sixty-four, of 250 H.P. and upwards each, are severally connected by crooked horizontal pipes with the chamber of a check-valve, with gag screw to make the same a stop-valve, such check-valve discharging into the side of a main steam-pipe running horizontally, with a slight drop toward vertical steam drums. These drums are made annular. The main steam-pipe to the street connects with the central cylinder, and the annular portion at the bottom collects water, which is withdrawn regularly by a trap. The steam enters the drum through tangential pipes, which cause it to circulate around the interior of the outer shell, and openings are provided to the interior shell below and at a considerable angle from the inlets. It will be seen that these drums are practically "separators," and they certainly have done their work very well. The pipes of the New York Steam Company were ordinarily laid to rise quickly and then run on a gradual descent away from the boiler house, with traps at the lowest point. In laying a 10-inch pipe on Madison Avenue, north of Fifty-eighth Street, it was necessary, however, to have an ascending grade for a long distance, and it was hoped that the water would run back under the steam to the boiler house at first, when there were but few customers connected to the pipe, and that afterward it would be distributed to the various buildings so as to cause no difficulty; but the slight current caused by a few customers held up the water so that it would collect in considerable quantities until it actually filled the pipes on the side of the streets, and the heating apparatus of the buildings connected thereto. By locating only two traps on the slope in a distance of about a quarter of a mile, the water carried forward by the steam was drained out sufficiently at intervals, so that all difficulties were overcome.

Recurring to the paper, Mr. McBride is entitled to great credit for his energy in tracing the difficulty to its source, and devising such simple means to overcome the same.

DCI.*

HEAT UNITS AND SPECIFICATIONS FOR PUMPING ENGINES.

BY ALBERT F. HALL, BOSTON, MASS.
(Member of the Society.)

THE writer's experience in connection with the *duties* of pumping engines has shown him that misunderstandings arise as to the meaning of a test based upon heat units, and he hopes that an explanation will not be unwelcome to those who have not given the matter special study.

In all measurements of comparison it is essential that a fixed and definite unit be adopted. The steam engine is a *heat engine*, and the unit adopted for the measurement of its performance is rightly taken as the *unit of heat*, and the number of foot-pounds of work performed per stated number of such units as the efficiency of the engine. It will readily be seen that coal can never be taken as a standard unit, as it is of a variable nature, and, even with any particular kind of coal, there are so many things connected with its use as to render it unfit to be chosen for measurement of comparison.

What is a *unit of heat?* This may be defined, according to the most modern views, as the quantity of heat required to raise the temperature of one pound of pure water from 62° to 63° Fahr. It is sometimes called a British thermal unit and abbreviated B. T. U.

It may be asked why the range of temperature is taken from 62° to 63°, instead of the usual range from 32° to 33°. In reply to this we cannot do better than quote from Professor Peabody's remarks in his book of tables of steam and other vapors: "Nothing is known about the specific heat of water from 32° to 33° Fahr.; consequently the commonly accepted value of the thermal unit is an ideal quantity inferred from the behavior of water at higher temperatures. It is more

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

scientific to take an easily verified quantity for the standard; and there is practical convenience in choosing 62° Fahr. for the standard temperature, because it is near the mean temperature of the air during experimental work."

The temperature of water at its maximum density has also been taken as the standard, but the same remarks made about the specific heat of water from 32° to 33° Fahr. also apply to this case.

Professor Rowland's recent carefully conducted experiments have shown the *unit of heat* to be equivalent to the work of raising 778 lbs. one foot high or one pound 778 feet high. This work of 778 foot-pounds is called the *mechanical equivalent of heat*, and is usually designated by the letter J, from the name of Joule, who made many extensive experiments relating to it. The reciprocal of J, or 1 divided by 778, is designated by the letter A.

Why should we not take a pound of steam for our unit? Because the heat required for the generation of steam varies with the pressure, and it requires more heat to convert water of a given temperature into steam of 200 lbs. pressure than into steam of 100 lbs. pressure. The difference is not great, but it is enough to make such a unit of little value in accurate experimental work. For ordinary work it is sufficiently accurate. But why use such variable quantities as coal and steam for units when the *heat unit* is just as simple to obtain and use, and one which requires but little study to understand? It is certainly preferable to have a *standard*, about which there can be little dispute, and such points as have arisen are of such a trivial nature as to have no effect upon the practical question of the use of the *heat unit* as this standard.

In order to have a clear idea of the practical use of this unit let us briefly study the generation of steam. The operation of the formation of steam from water of a given temperature may be divided into three stages:

First, Raising the temperature of the water to that corresponding to the pressure of the steam.

Second, Converting the water from this point into steam.

Third, Heating the steam and increasing its volume above that corresponding to its pressure at saturation, or what is commonly called superheating it.

During the first stage, the heat which has been given to the

water is more than that indicated by the thermometer, a small additional quantity, sometimes called latent, being required to obtain this temperature, that shown by the thermometer being called sensible.

During the second stage, the amount of heat required is that necessary to overcome internal and external resistances.

During the third stage, the heat is used to raise the temperature of the steam and increase its volume, its pressure remaining *constant*.

In tables of the properties of steam all quantities of heat are usually reckoned from the freezing point, or 32° Fahr., and the various stages are divided as follows:

The quantity of heat necessary to raise one pound of water of the temperature of 32° Fahr. to the temperature of the steam of a given pressure is called the *heat of the liquid*, and is usually designated by the letter q, the temperature being reckoned from 0° Fahr. and indicated by t.

The heat used during the second stage is called the *heat of vaporization*, and is designated by r. This heat of vaporization is divided into two parts—that required to produce internal changes, and that for overcoming the external pressure or for performing external work. The first quantity is represented by the Greek letter ρ (rho), and the second by $A \times p \times u$—in which A represents the reciprocal of the mechanical equivalent of heat, as already explained; p the pressure in pounds per square foot; and u the increase in volume, in cubic feet, above that which the water had at the end of the first stage. $p \times u$ equals the number of foot-pounds of work done against the external resistance, in increasing the volume which the water had at the end of the first stage to that which it has when converted into steam. Now, if we divide this amount of work by the foot-pounds of work equivalent to one unit of heat, or, what is the same thing, multiply it by its reciprocal, we shall obtain the *heat equivalent* of the external work.

The heat used during the third stage is called the specific heat of steam under *constant pressure*, and is usually designated by c_p. Regnault found that for every degree Fahrenheit that steam is superheated under constant pressure, 0.4805 of a thermal unit is required.

The total quantity of heat consumed during the first two stages, or that required to convert one pound of water of the

temperature of 32° Fahr. into steam of any given pressure, is usually expressed by the Greek letter λ (lambda), and we have, consequently,

$$\lambda = q + r = q + (\rho + Apu).$$

Suppose, for example, that we have steam of 150 lbs. pressure per square inch above zero, or about 135 lbs. per square inch above the atmosphere, as shown by the gauge. The total heat required to convert one pound of water of the temperature of 32° Fahr. into steam of 150 lbs. absolute pressure per square inch is given in Professor Peabody's tables as

$$\lambda = 1191.2 \text{ B. T. U.}$$

If we assume that the water at its entrance to the boiler had a temperature of 98° Fahr., then the heat required to raise water of 32° Fahr. to this temperature would be:

$$q = 66.01 \text{ B. T. U.}$$

Consequently

1191.2 less 66.01, or 1125.19 B. T. U.,

have been given to every pound of water converted into steam of the pressure of 150 lbs. per square inch above zero.

Let us also assume that calorimetric experiment has shown that only 98% of the water was converted into steam, or, in other words, that the steam contained 2% of moisture. In this case we have a mixture of steam and water, and must, consequently, make allowances, since less heat has been used per pound of water than that already obtained. The heat of vaporization of steam of this pressure is given in the tables as:

$$r = 861.2 \text{ B. T. U.}$$

Since 0.98 of a pound of water was converted into steam, then

0.98 × 861.2, or 843.976 B. T. U.,

were absorbed by that part of the one pound of water which was converted into steam. The heat required to raise the

temperature of water of 32° Fahr. to that corresponding to steam of the pressure of 150 lbs. per square inch above zero is

$$q = 330 \text{ B. T. U.}$$

And as *all* the water received this amount of heat, reckoned from 32° Fahr., then

$$843.976 + 330, \text{ or } 1173.976 \text{ B. T. U.},$$

are the units of heat required to raise one pound of water from a temperature of 32° Fahr. to the temperature corresponding to the pressure of 150 lbs. per square inch above zero, and to convert $\frac{98}{100}$ of a pound of water into steam of this pressure.

But, as the initial temperature of the water at its entrance to the boiler was 98° Fahr., and the heat of this water, reckoned from 32° Fahr., was, as we have seen, 66.01 B. T. U., we have the total quantity of heat imparted to the pound mixture of water and steam, as

$$1173.976 - 66.01 = 1107.966 \text{ B. T. U.},$$

instead of

$$1125.19 \text{ B. T. U.},$$

required to convert all of the water into steam.

The understanding of this simple example is about all that is required fully to appreciate the calculations based upon heat units. There are, of course, allowances to be made in connection with various apparatus used; but such considerations belong rather to a thorough understanding of the apparatus than to an understanding of the calculations as such, and the Transactions contain ample examples of such applications in the report of the committee on duty trials.

In determining the quantity of heat used in the jackets, care must be taken to avoid a common error. The steam in the jackets is condensed under *constant pressure*, and the heat given up by each pound of this steam is the total heat reckoned from 32° Fahr. required to generate steam of the jacket pressure, less the final heat of the water as it leaves the jacket, and not, as some have put it, this amount less the heat $A \times p \times u$ required for external work.

Having now discussed the measurement of the heat used in the steam, we shall touch upon the question of DUTY.

The boiler and the engine should each be tested independently, for the reason that a good engine should not suffer from a poor boiler, or *vice versa*, and because, as these are often furnished by different makers, the merits of each ought to be known.

The rating of the "duty" of pumping engines upon the number of foot-pounds of work performed per 1,000,000 *heat units* was first suggested by Dr. Charles E. Emery,* member of this Society, and this unit has been very wisely recommended by a committee appointed by this Society to investigate the matter. It has in it the elements of simplicity, and places all engines upon an equal footing for comparison, which no other proposed method does.

It must not be forgotten that *coal* has nothing to do with this basis.

We are now prepared to examine two cases recently published in the specifications for pumping engines for two of our large cities, and to point out the errors. More might be cited, but these will be sufficient as examples. The first case is as follows:

"The duty required of the engine will not be less than one hundred and twenty million (120,000,000) of foot-pounds, with one hundred and forty (140) pounds of steam pressure at the engine throttle.

"The equivalent of one hundred (100) pounds of coal being taken as one million (1,000,000) heat units."

"The total number of heat units shall be measured from the temperature of the feed-water taken at a point where it enters the boiler, to steam of the temperature shown by pressure gauges on the boiler.

Nothing is said in the opening paragraph about the basis upon which the duty is to be made, and the amount of duty and allowable steam-pressure are spoken of in an unsatisfactory manner.

The statement is then made that the equivalent of 100 lbs. of *coal* will be taken as 1,000,000 heat units, which indicates that coal is to be estimated from the number of heat units used, the logic of which is not apparent, since the duty is really based upon heat, as will be shown by the formula given for its calculation.

* Page 21, Vol. VI., U. S. Centennial Commission International Exhibition, 1876.

There is not only an inconsistency as to the basis of measurement, but an error in the statement that the heat is to be measured to steam of the temperature shown by pressure gauges. Temperatures are not shown by *pressure* gauges. Compound gauges have been made where the temperatures corresponding to the pressures are given, but this is unusual, and as no such gauges are required by the specifications the statement must be considered as somewhat careless.

The formula for duty is as follows.

$$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\text{Total number of heat units consumed}} \times 1{,}000{,}000.$$

Now, it is easily seen that the matter of coal does not enter into the above computation, and the use of the expression is entirely unwarranted and absurd; the more so, since nothing is said in the specifications about weighing the coal.

Had the requirements for duty been worded as follows, no ambiguity would exist, and all would be simple and logical:

The engine must perform the work of not less than one hundred and twenty million (120,000,000) foot-pounds per one million (1,000,000) units of heat when running with a steam pressure at the engine throttle of one hundred and forty (140) pounds above the atmosphere, the heat utilized to be measured as follows:

Each pound of water fed to the boiler is to be debited with the heat required to raise *all* of the water from the temperature it has at its entrance to the boiler to that corresponding to the boiler pressure, and the amount of heat required to convert 97% of the water into steam of boiler pressure from the temperature corresponding to this pressure.

The formula for duty as already quoted can then be given.

The reason why the above directions "for the determination of heat utilized" have been given is because there is a clause allowing three per cent. for entrained water in the steam without explaining how this allowance shall be made.

This wording for the calculation of heat is of vital importance in principle, bearing in mind that the steam engine is a *heat* engine. The determination of *dry steam*, as has been done in several instances, by deducting the moisture bodily from the

feed-water, when such moisture actually enters the cylinder, is entirely wrong. Although moist steam is considered detrimental to economy, if it enters the cylinder it carries with it *heat*, and in deducting such moisture we virtually debit the engine with less *heat* than it actually used, and the deduction has no *raison d'être*. Such a deduction also shows a lack of knowledge of the real nature of the case, since there is no known law by which the injury from such moisture can be calculated.

It has been suggested that it would be better to give the duty per ONE unit of heat instead of 1,000,000, and thus still further simplify the work. The idea of having a basis which would be somewhat near the old coal basis was a good one, but we could easily become accustomed to the other.

Another case will now be cited. The specifications say:

"The contractor must guarantee that the engine shall develop a duty of a certain number of million foot-pounds, based upon an evaporation of ten pounds of water from and at 212° Fahr. per pound of coal."

This statement is very clumsy, and not only another misapplication of the use of coal, but another basis for calculating duty.

To convert one pound of water of the temperature of 212° Fahr. into steam of the same temperature, we require 965.8 B. T. U.

If we have used w pounds of feed-water, and have given to every pound of this water H heat units, then

$$w \times H$$

is the total quantity of heat used. The specifications ask us to ascertain how many pounds of water of the temperature of 212° Fahr. would have been evaporated into steam of the temperature of 212° Fahr., with the same expenditure of *heat*, which is found by dividing

$$w \times H \text{ by } 965.8.$$

It then tells us to divide this by 10 and call it coal (it might as well have said straw). We are then directed to divide the foot-pounds of work done by this fictitious amount of *coal* and to multiply the result by 100, the final result being the duty obtained. In algebraic language this is simply:

$$\text{Duty} = \frac{\text{Foot-pounds of work done}}{\frac{w \times H}{965.8} \div 10} \times 100$$

$$= \frac{1000 \times 965.8 \times \text{foot-pounds}}{w \times H \text{ (or heat used)}}$$

$$= \frac{965{,}800 \times \text{foot-pounds}}{w \times H},$$

which, in simple language, is merely the duty in foot-pounds of work done per 965,800 units of heat.

What excuse can be given for basing the duty upon 965,800 units of heat, when that based upon 1,000,000 would involve less work in calculation, besides being in harmony with modern ideas?

It is hard for those not acquainted with the true nature of this matter to avoid the use of the *word* coal (coal itself does not enter into the requirements), and the object of this paper is to make the matter clear.

The writer would certainly advocate the appointment of a committee whose duty it shall be to frame a series of forms to be inserted in specifications for pumping plants to fit the various conditions likely to occur in practice.

These forms should not only be accurately worded from a technical standpoint, but should be expressed in simple and correct English, to avoid all misunderstandings.

Although the writer does not advocate the testing of pumping engines upon a *coal* basis, yet, as many do not hold the same views, it would be well to include *proper* forms for such a basis.

For convenience of reference is appended the following

SUMMARY OF TERMS USED IN STEAM TABLES.

J = 778 foot pounds = mechanical equivalent of heat.
A = reciprocal of J, or 1 divided by 778.
p = pressure above zero in pounds per square foot.
u = difference between the volume of one pound of water of a given temperature and that of one pound of steam of the same temperature, in cubic feet.
$t°$ = temperature in degrees Fahrenheit.

q = heat of the liquid, or the amount of heat required to raise one pound of pure water from the temperature of 32° Fahr. to a given temperature.

r = heat equivalent of vaporization, or the amount of heat required to convert one pound of pure water of a given temperature into steam of the same temperature.

ρ = heat equivalent of internal work of vaporization, or that part of the heat r required for internal changes.

Apu = heat equivalent of external work of vaporization, or that part of the heat r required for external work.

$r = \rho + Apu$.

$\lambda = q + r = q + \rho + Apu$ = the total amount of heat required to convert one pound of pure water from the temperature of 32° Fahr. into saturated steam of a given temperature.

$c_p = 0.4805$ = specific heat of superheated steam of constant pressure, or the amount of heat required to raise one pound of superheated steam of constant pressure through one degree Fahrenheit.

DISCUSSION.

Mr. F. R. Hutton.—It will be noticed that the author, upon the sixth page of his paper, credits the work of Dr. Chas. E. Emery, member of this Society, with the first suggestion of the heat-unit basis as a means of determining engine efficiencies. Some interesting and creditable correspondence has come under my hand, from the Chairman of the Committee of this Society, on a standard method of conducting duty trials of pumping engines, in which he expresses to Mr. Emery his regret that he had not known of this earlier suggestion of the heat-unit basis, that he might have given proper credit when this method was recommended by that committee. Mr. Emery in reply gracefully disclaims any occasion for regret, and points to the original reference, pages 21 and 115 of the report of the judges of Group XX. of the Centennial Exhibition.

Inasmuch as a table on page 121 of that report shows the relative efficiencies of different kinds of engines compared on the same basis, I desire to append that table in the discussion of this paper, especially as those reports are now becoming difficult of access, and the table is a most interesting standard for comparative reference.

TABLE OF THE EFFICIENCIES OF STEAM-ENGINES COMPARED ON THE BASIS OF THE COST IN THERMAL UNITS, BY CHARLES E. EMERY, PH.D., AS PUBLISHED AT PAGE 121 IN THE REPORT OF THE JUDGES OF GROUP XX., CENTENNIAL EXHIBITION (LIPPINCOTT & CO., 1878), WITH LINES 2, 3, AND 4 BROUGHT DOWN TO DATE, AND FULLER EXPLANATORY HEADINGS AS STATED IN TEXT.

1		2	3	4	5	6	7	8	9
Number of Line for Reference.	Description.	Steam Pressure.	Ratio of Expansion.	Water Evaporated per Indicated Horse-power per Hour.	Duty in Foot-Pounds per Thermal Unit, or in Millions of Foot-Pounds per Million Thermal Units, or in Millions of Foot-Pounds per 100 lbs. of Coal, with a Calorific Value in Practice of 10,000 Thermal Units per Pound.	Efficiency of Engine based on the Proportion of the Heat in the Steam above 100° Fahr. utilized in the Engine.	Efficiency of Engine and Boiler based on the Proportion of the Heat in the Steam above 100° utilized in the Engine when the Steam is furnished by a Boiler of substantially 69% Efficiency, in which 10,000 Thermal Units are Imparted to the Water per Pound of Coal.	Net Effective Duty for lifting Water, in Millions of Foot-Pounds, per 100 lbs. of Coal, with a Calorific Value in Practice of 10,000 Thermal Units per Pound.	Thermal Units per Horse-power per Minute.‖
	Calculated performance, maximum	.....	.....	.....	772	1.00	0.69	.....	42.7
	" " "	.....	.....	.....	778, lines 2, 3, 4	.....	.....	.....	42.4
2	Calculated performance*	120	20	7.315	241.5	.310	.214	.....	136.7
3	Milwaukee pumping engine (1891)*	121.6	19.55	11.8	151.6	.195	.134	137.1	217.7
4	Newton Cross compound pumping engine. Geo. F. Blake Mfg. Co. builders*	122.3	18.3	14.07	119.4	.153	.106	.....	276.4
5	U.S. revenue steamer *Rush*,† compound engine	69.2	6.22	18.384	97.03	.126	.087	.....	340.1
6	" " " *Gallatin*,† vertical cylinder with steam-jacket	67.2	4.19	21.48	83.08	.108	.074	.....	397.2
7	" " " *Dexter*,† vertical cylinder without steam-jacket	67.1	3.49	23.905	74.66	.097	.067	.....	412.0
8	" " " *Dallas*,† vertical cylinder without steam-jacket	32.0	3.13	26.945	66.91	.087	.060	.....	493.2
9	" steamer *Mackinaw*,‡ inclined cylinder without jacket	49.0	2.2	30.306	59.16	.077	.053	.....	557.8
10	" " *Mackinaw*, steam superheated	52.0	3.2	22.725	78.83	.102	.070	.....	418.6
	Non-condensing engine, with governor cut-off§ (st. jacket)	81.7	5.0	25.482	69.81	.090	.062	.....	472.7
12	Non-condensing engine, regulated by throttle	.....	.....	.....	30 to 45	0.4 to .06	.03 to .04	.....	.....
13	Pumping engines							30 to 137*	.....
14	Steam-pumps, large size, proportioned for the work to be done							15 to 30	.....
15	Steam-pumps, small sizes for ordinary uses. See page 22, Report							8 to 15	.....
16	Vacuum-pumps. See page 21, Report							3 to 10	.....
17	Injectors, when used for lifting water, not required to be heated. See page 96, Report							2 to 5	.....

* Modified. See text.

† See article by the writer on "Compound and Non-Compound Engines," *Transactions American Society Civil Engineers*, Vol. III., p. 68, 1875; *Journal of the Franklin Institute*, February and March, 1875; *Engineering* (London), January, February, and March, 1875; *Proceedings of Institution of Civil Engineers* (British), Vol. XLI., p. 292, and Vol. XLI., p. 296; also report of trial of United States revenue steamer *Gallatin*, *Journal of the Franklin Institute*, February, 1876, and Vol. XXI., *Engineering*, 1876.

‡ See Vol. II., *Isherwood's Experimental Researches in Steam Engineering*, pp. 77-116.

§ *American Institute Reports*, 1869-70, 1870-71.

‖ This column was not in the original table, but has been added at the request of Mr. Hall.

While I had at first thought that it would be necessary to reproduce a large portion of the text in order that the table would be understood, in conference with Mr. Emery it has been decided slightly to elaborate the headings of the table so as to include explanations clearly stated in the text. In doing this, opportunity has been taken to modify lines 2, 3, and 4 so that they will refer to the performance on the same basis of the Milwaukee pumping engine, built by the E. P. Allis Company from the designs of Mr. Reynolds, reported by Professor Thurston at the New York meeting, December, 1893, and the performance of the Newton (Mass.) pumping engine, reported by Mr. Dean, thus literally bringing the upper part of the table, showing the higher efficiencies, down to date.

With the table thus modified before us, we may examine what it really shows. In the first place, on pages 21 and 115 of the report Mr. Emery not only made his comparison on the basis of foot-pounds per thermal unit, but suggested that the result be stated in the evidently equivalent expression millions of foot-pounds per million thermal units, or per 100 lbs. of coal, with a calorific value in practice of 10,000 thermal units per pound, which he points out, equals very nearly the evaporation of 9 lbs. of water at a pressure of 80 lbs. from a temperature of 100°. Referring to column six, line three of the table, it will be seen that even the most economical engine so far reported does not quite reach an efficiency of 20%, and that the efficiency of the engine and boiler combined, for practically 69% efficiency of boiler, is only 13.4%, or but little more than one-eighth the calorific value of the fuel. This is on the basis that the ultimate calorific value of the fuel is 14,500 thermal units (see page 74 of report). Mr. Emery has kindly modified line two to show the highest performance possible for the steam pressure and ratio of expansion used during the test of the Milwaukee engine referred to in line three, but wishes it understood that line two really shows a somewhat higher efficiency than the conditions warrant; for, although the water condensed to furnish the heat expended in the performance of work was included in making up the weight of feed-water, the mean pressure was, to save time, worked out by the usual hyperbolic formula. We would suggest that Mr. Emery could very properly make this feature the subject of a special paper at a subsequent meeting. It will be observed that the efficiencies are worked out on the basis that the thermal equivalent of heat is

772 foot-pounds, which was the best accepted value at the time the article was originally written. The value of 778 foot-pounds has, however, been used in calculating lines 2, 3, and 4.

After submitting the above for Mr. Hall's closure we have acceded to his request to add column nine to the table, the quantities in which evidently equal 33,000 divided by those in column five, and also accede to his further request to republish pages 20–22 from the original report which he declares very interesting. Those who have access to the report will find pages 115 to 121, which include the table herewith presented, also very instructive and interesting.

Duty of Pumps.

EXTRACT FROM GENERAL REPORT OF THE JUDGES OF GROUP XX., CENTENNIAL EXHIBITION, WRITTEN BY CHARLES E. EMERY.

"The duty of a vacuum pump may be readily ascertained by simply noting the height of lift, and the initial and final temperatures of the water lifted. All the heat of the steam not expended in work enters the water, and the work performed lifts the same water. The difference in temperature gives very nearly the number of heat units imparted to each pound of water lifted, and each pound of water so heated is lifted a certain number of feet high, so the result may be expressed readily in foot-pounds per heat unit, and this may be changed into terms of the conventional duty of pumping engines by simply fixing upon a standard to represent the number of heat units imparted to water by one pound of coal. Inasmuch as the duty of pumping engines is conventionally expressed in the number of foot-pounds obtained by the consumption of 100 lbs. of coal, and as a pound of coal in a good boiler will evaporate over 9 lbs. of water at such pressure as to impart 10,000 heat units to the water, or 1,000,000 heat units for 100 lbs. of coal, the writer has proposed the latter as a convenient basis of comparison, since the number of foot-pounds per heat unit evidently expresses also the duty in millions of foot-pounds per 100 lbs. of coal. For ordinary comparisons, the number of millions duty equals the lift divided by the difference between the initial and final temperatures of the water. For more accurate computation, the divisor should be increased by the number of heat units expended for work per pound of water lifted, which equals the height divided by 772. The height preferably should be calculated from the indications of a pressure-gauge at the bottom of the discharge pipe, so as to include

frictional resistances. If D = duty in foot-pounds per 100 lbs. of coal, H = the height of lift per gauge, and t and T = the initial and final temperatures respectively, then

$$D = \frac{1{,}000{,}000\ H}{T - t + .0013\ H}$$

"Arrangements have been made by the writer to use the same basis in testing pumping engines by discharging water from the hot well into the suction of the main pumps and noting the resulting increase of temperature with delicate thermometers.

"A vacuum pump tested by the writer in 1871 gave a duty, on the above basis, of $4\frac{7}{10}$ millions; one tested by Mr. J. F. Flagg at the Cincinnati Exhibition in 1875, reduced to the same basis, gave a maximum duty of $3\frac{25}{100}$ millions. Several vacuum and steam pumps tested on this basis at the suggestion of the writer about two years since, gave duties reported as high as 10,000,000 to 11,000,000, the very small steam-pumps doing no better apparently than the vacuum-pumps, which is by no means surprising. Elaborate experiments made with steam-pumps at the American Institute Exhibition of 1867 * showed that average-sized steam-pumps do not, on the average, utilize more than 50% of the indicated power in the steam-cylinders, the remainder being absorbed in the friction of the engine, but more particularly in the passage of the water through the pump. Again, all ordinary steam-pumps for miscellaneous uses require that the steam-cylinder shall have three to four times the area of the water-cylinder to give sufficient power when the steam is accidentally low; hence, as such pumps usually work against the atmospheric pressure, the net or effective pressure forms a small percentage of the total pressure, which, with the large extent of radiating surface exposed and the total absence of expansion, makes the expenditure of steam very large. One pump tested by the writer required 120 lbs. weight of steam per indicated horse-power per hour, and it is believed that the cost will rarely fall below 60 lbs.; and as only 50% of the indicated power is utilized, it may be safely stated that ordinary steam-pumps rarely require less than 120 lbs. of steam per hour for each horse-power utilized in raising water, equivalent to a duty of only 15,000,000 foot-pounds per 100 lbs. of coal on the same basis adopted for the vacuum pumps. With larger steam-pumps, par-

* See report of Messrs. Holmes, Selden, and Emery, Judges, etc., *Transactions American Institute*, 1867-68.

ticularly when they are proportioned for the work to be done, the duty will be materially increased."

Mr. Albert A. Cary.—The use of heat units in specifications is by no means confined to specifications for pumping engines alone, as we find, within the past year, a growing tendency among engineers to call for guarantees of efficiency in boilers, in which those bidding are to state the percentage of the total number of heat units contained in the coal which they will utilize in the generation of steam, in their particular boiler.

The report of this Society's committee on "A Standard Method of Conducting Locomotive Tests," in which the committee recommend the use of "a standard coal showing 12,500 B. T. U. per pound,"* to allow a fair comparison to be made in the relative efficiency of different locomotives, tested at different times, and besides the interesting paper read at the Chicago meeting † by Mr. Barrus, describing the coal calorimeter used by him and its field of use, seem to have stimulated engineers to the use of the heat unit in their specifications, as, prior to the time of the valuable papers referred to above, boiler makers were never called upon to make such guarantees.

During the past year I can recall, among others, two specifications, one for boilers to be used in a pumping station in St. Louis, where bidders were called on to guarantee 60% efficiency for their boilers, and the other specification came from Philadelphia, in which the bidders were asked to state the efficiency they would guarantee.

In the first case—the St. Louis plant—a certain bonus or premium was offered for each per cent. above the 60% efficiency the boiler might develop.

This, I certainly think, is a step in the right direction.

The old way of guaranteeing a boiler, based on its evaporation per pound of coal, when we do not know how much heat is held in the coal, is certainly very indefinite and unsatisfactory, and what do we know, except, maybe, by some guessing comparison, about the real efficiency of the boiler that has just been tested?

As was rightfully stated in the committee's report on locomotive tests, "It is fully as important to know the character of the fuel as to find the number of pounds of coal burned, and no test can be satisfactory that leaves out the heat value of the fuel."

The fuel calorimeter has now furnished us with a comparatively

* See *Transactions*, Vol. XIV., p. 1317. † See *Transactions*, Vol. XIV., p. 816.

simple method for determining the heat units liberated in the combustion of coal, and Mr. Barrus gave us the description of an excellent form of this instrument at the Chicago meeting above referred to.

Experience has proved that this method of determining the heat value of fuel is not only, by far, simpler, but also more reliable than that of making a chemical analysis from which to calculate the theoretical calorific value of the fuel, as we are not yet wise enough to predict the *exact* combinations which will be made in the process of combining the various elements we are bringing together to accomplish combustion, and it probably is for this reason that we find in so many specifications the statement that the calorific value of the fuel is to be determined by the fuel calorimeter.

Before closing these remarks it might be well to state that the efficiency of the same boiler will be found to differ as different fuels are used. Thus it has been found that a certain boiler will utilize over 80% of the calorific value of the fuel when petroleum oil or a high-grade anthracite coal is used, but when some of the low-grade Western coals are used, it is found impossible for this same boiler to utilize more than from 40% to 50% of the calorific value of the fuel.

This difference does not apply to any one type of boiler; and, therefore, it is not attributable to boiler design, but is rather attributable to a lack of knowledge on the part of engineers, who should study this subject carefully, and by systematic experimenting make this fuel yield up a proper percentage of its heat in the generation of steam.

*Mr. Albert F. Hall.**—The writer desires to thank both Professor Hutton and Dr. Emery for the interest they have shown in adding to the value of his paper by their friendly contributions.

It is to be hoped that he will not seem to be claiming more than his just dues in adding that among those who responded to an invitation for an expression of views upon duty trials, the writer alone suggested the heat unit as a standard, and that it was not until after much persuasion on his part that it was adopted.

It is a pleasure to note Mr. Cary's remarks and to learn that there is a growing tendency to use the heat unit as a standard, and also to note that the requirements for a standard coal to be used in locomotive tests put these tests upon a heat-unit basis.

* Author's Closure, under the Rules.

DCII.*

A NEW RECORDING PRESSURE GAUGE FOR EXTREMELY HIGH RANGES OF PRESSURES.

BY WM. H. BRISTOL, HOBOKEN, NEW JERSEY.
(Member of the Society.)

THE gauge herein described and illustrated, although it may be adapted to ordinary ranges, has been designed especially for high ranges of pressures and to supplement the sinuous tube and diaphragm forms of recording gauges previously described.† The principal object has been to produce a gauge which would withstand working pressures of 1,000 lbs. or more, such as are employed on oil pipe lines and hydraulic press work, without taking permanent sets, and one in which the readings for increasing and decreasing pressures would be the same. The writer has had difficulty in finding an ordinary reading gauge for high ranges of pressure which has been at all reliable in these respects. In testing and standardizing high pressure gauges the most satisfactory results have been obtained with dead weights carried by a cap covering an exact square inch of area. At the Stevens Institute for test work, where high pressures are involved, a cylinder and plug of accurately known cross section has been frequently employed on account of difficulty to secure reliable high pressure gauges.

In Fig. 327 is shown the interior construction of a new recording gauge in which readings agree perfectly for increasing and decreasing pressures, over the entire scale. The spring part of the gauge consists simply of a Bourdon tube of flattened cross section, wound into a small helical form of four complete revolutions. One end of the tube is rigidly fastened to a bracket with an opening for the pipe communicating to the gauge; and on the end, free to move by application of pressure, a pen arm *B* is directly secured. The diameter of the helical coil being only one inch,

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† *Transactions American Society of Mechanical Engineers*, Vol. XI., p. 225, No. 368; and Vol. XIV., p. 325, No. 518.

the actual motion of the free end of the tube is small, and as this motion is distributed through the four complete turns of the tube, it is evident that in the angular movement necessary to carry the pen arm over the scale of the chart, the tube itself is not strained to a point approaching its elastic limit. By this construction, the necessity of multiplying devices is entirely avoided. By varying

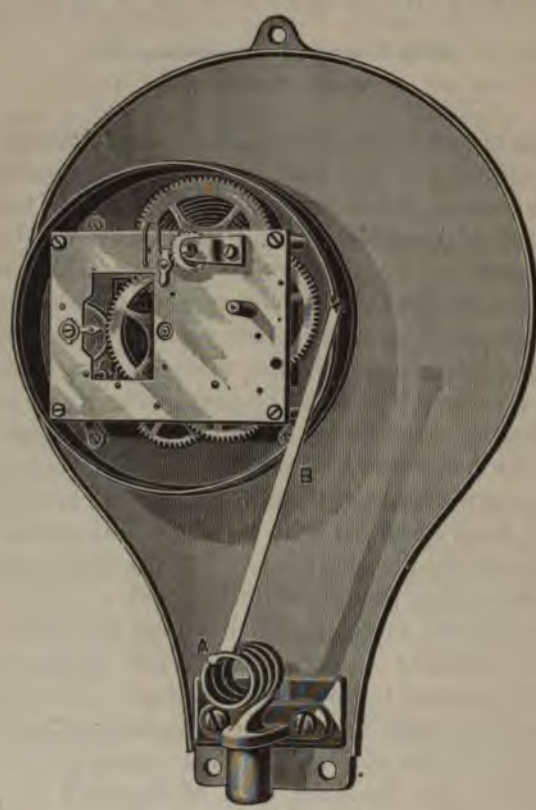

FIG. 327.

the thickness of metal the cross section and the number of revolutions of the tube, a helix adapted to any desired range of pressure may be obtained even as low as five pounds per square inch for the whole scale. In the manufacture of the helical tube there is no difficulty in reproducing them to fit a standard chart, as illustrated in Fig. 328, which shows a complete gauge ready for application.

In this instrument, as in forms of recording gauges previously described, the pen which is directly attached to the helical tube records with ink on a chart revolved by a clock movement once in twenty-four hours. In Fig. 328 the total range is 1,500 lbs. per square inch.

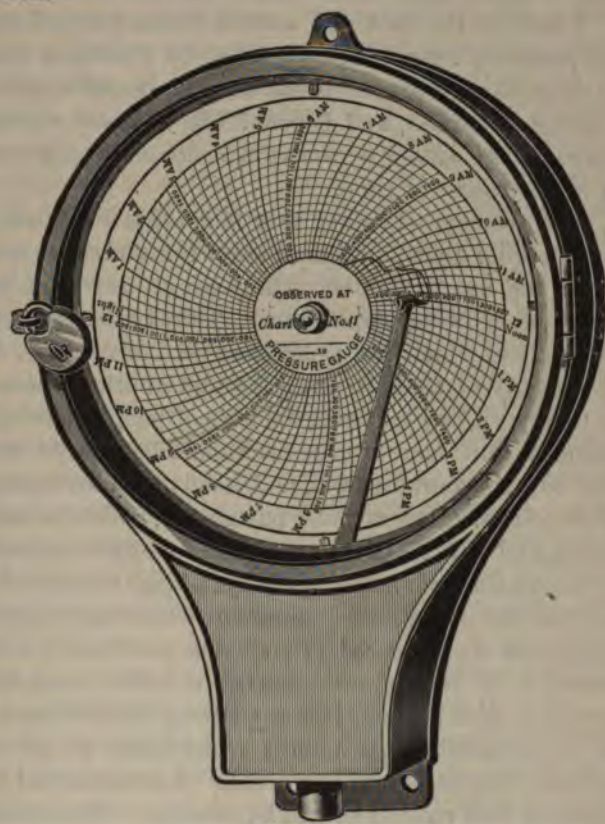

FIG. 328.

This gauge is the joint invention of Mr. E. H. Bristol and the writer.

DISCUSSION.

Mr. Gustavus C. Henning.—This is a third kind of gauge presented before the Society by Professor Bristol, and I have asked the question once before whether these devices take any change of form during consecutive periods of action. This is a form which is more likely to suffer from action on account of the mass of material that must be in the tube to withstand the pressure, and I

am inclined to believe, although I have not any experimental data except the action of the common Bourdon tube, that there will be a gradual change of the action in the tube. I would like to ask whether any member has any knowledge from experimental observation whether these tubes remain constant or whether they change. I believe the first and second forms for ordinary and very light pressures which were described by Professor Bristol do not show these changes or variations like the Bourdon tube. This is a new thing, of course, and I would like to know whether this heavier tube under heavier pressures will not show slight changes of rating in the course of time.

Mr. J. F. Holloway.—I regret very much that Professor Bristol is not here, in order that he might answer this and other questions. I look upon the use of the recording gauge as one of growing interest and of growing importance. The constantly increasing number of boilers which are being used in our country, and the fact that so many of them in our cities and in other places are so placed as to imperil, as I may say, the lives of people, I think render it very essential that some apparatus should be devised and used, which will be a constant, reliable, and steady monitor upon the action of such boilers. Boilers under pavements, boilers in vast apartment houses, and boilers on steamboats, locomotives, schoolhouses, etc., have above and around them always a large number of people. They are often not cared for to the extent which they ought to be, because they are often located in positions not readily accessible. Their record is of very great importance, and I think the Society does well to give attention to any efforts that are made in the direction of producing instruments which shall be an accurate record of what is going on within them. But above all things it is essential that such records should be accurate. When we depend upon any piece of mechanism to do work of this kind, it should be of such a construction that we feel perfect confidence in the record it makes. A record which is not true is worse than no record, because we become accustomed to rely upon it. Any effort which may be made toward the production of any instrument which shall constantly produce an accurate result is one that we should all welcome and should be glad to see brought forward and come into very general use.

The question which Mr. Henning has asked is a question that was in my mind as well, as to whether this particular mode of producing the record is one which will be reliable, and one which

will be as good five or ten years from now as when it is put in, and I simply repeat that if it fails to be continuously accurate, it fails of the purpose for which it should be used.

I would very much like to know what effect would be produced by the constant use of a Bourdon spring to make such a record.

There is one other feature. Springs of all kinds in instruments attached to boilers are subject to great changes of temperature. There are times, of course, when the steam from the boiler will impinge on the mechanism which is to resist the pressure, and in this case, of course, with this additional long-coiled spring, there is more surface to be affected by this difference in temperature. Just what effect that will have on the spring, whether the spring will produce one result when the steam is in it, and another result when the water is in it, is another question I would like to ask.

This, in many respects, is similar to a gauge, I think, that was described in a paper read before the Society at some former period. I do not remember what was said about it at that time, but I would also have liked to ask what will be the effect of friction on the mechanism which produces the time results, the clockwork of the apparatus, when in some circumstances the pencil which marks the diagram is near the centre of its motion, and, of course, subjects the moving diagram to the least action, as compared to other cases, when the pencil moves out on a larger diameter, and which would, as it seems to me, increase the friction of making the diagram itself. Of course, in any mechanism which is moved by delicate clockwork we can understand that a difference in friction must make a difference in results which ought not to exist.

Another feature which I think is very essential in an apparatus of this kind is, that it should be of such a nature as to be unaffected by the jars and shocks which are incident to all apparatus connected with steam-engines and with steam-boilers, and that no movement of the apparatus produced by any shock should affect seriously the diagram which it makes. In short, the truthfulness of the thing is the foundation of it all. When it ceases to be truthful it ceases to be useful.

The matter of reading the diagram is perhaps a little complicated, but in that respect anybody would soon become accustomed to it, although we are most accustomed to reading diagrams which are in parallel lines.

I regret very much that the author of the paper is not present to help us in understanding some of these points.

Mr. F. H. Laforge.—I have had an extended experience in testing steam-gauges, and I may say that this gauge of Mr. Bristol's is simply an extension of the Bourdon spring principle, and, further, the Bourdon spring is not reliable. While I consider the Bourdon spring gauge as good if not better than any other spring gauge in the market, still I find that they do not continue accurate and reliable for any length of time. Almost universally the Bourdon spring weakens, and, consequently, the gauge indicates more than the actual pressure.

The error in this direction is much greater when the gauge is subjected to heat. As, for example, when the gauge is piped directly from the boiler with no siphon in the pipes, or when the gauge is fastened on the cast-iron front of a horizontal tubular boiler.

The Bourdon spring gauge when wrong is usually wrong to the same extent and in the same direction for all indicated pressures, or, in other words, if such a gauge indicates twenty-five pounds when the actual pressure is twenty pounds it will also indicate five pounds above the actual pressure at all points in the scale; this is not true of other spring gauges: when in error they indicate above or below the actual pressure and in varying amount.

Mr. A. A. Cary.—After a considerable experience in experimenting with springs made of different materials, I have found that one made of tempered steel stood up better under work, and retained its original shape and elasticity much better, than a spring made of any metal made hard by working and afterwards coiled or shaped into the required form. Most of the Bourdon springs are made of a tube of hard-drawn brass or similar metal, which tube is formed into the familiar shape, and, as you all know, they become affected sooner or later, and generally the accuracy of the gauge declines with its age.

When in the spring business, many hundreds of springs were brought to me, with orders to make other springs similar to them (in strength, size, etc.) of tempered steel. Thus, being obliged to test carefully springs made of nearly every metal and material used for that purpose, I had ample opportunity to compare the relative merits of all. During some of these experiments I took compression springs made of hard-drawn wire and several different metals, put them under a vibrator, which compressed and released them a great many times, and I found they

would always lose their spring-like qualities. After these experiments, I would replace these with similar, well made, tempered steel springs, and almost invariably I would find them as perfect as at the beginning, after thousands of compressions.

The only recording gauge I know of using a tempered steel spring is one made by Mr. Jarvis B. Edson of New York. In this instrument he has a circular plate of steel, forming a diaphragm which he marks or rather corrugates, in concentric circles. The pressure of the steam strikes the under side of the diaphragm. or spring, which is securely held around the outer edge, and causes it to rise in the centre, and its spring qualities cause it to fall as the pressure is reduced, and thus he gets practically a uniform motion, which varies directly with the pressure. I have used a gauge of this kind for a number of years, and found it working accurately after four or five years' service.

The explanation of this matter of the superiority of a tempered steel spring seems to be that, after the spring is formed into shape and is put into the fire for tempering, all the strain that is put on the metal when the spring is being formed is relieved by the heating process, and all the molecules of the metal assume a position of rest in relation to one another, and thus the metal contained in the spring is relieved of all internal strain when the spring is at rest and doing no work. To illustrate this point, after coiling a spring around an arbor, you will see, after a moment's reflection, that the outer portion is under a strain of tension and the inside of the spring is under a strain of compression.

Now, should this spring be used, still carrying these strains, and have the wire composing it subjected to the additional strains of torsion and bending (which it resists when in use), or should the wire be relieved of the strains put upon it in the process of manufacture before being subjected to its natural strains when in use? Surely, there can be but one answer to this.

This strained and re-strained condition is just the one found in all the springs made of metals which are made hard by working and then formed into springs, depending entirely on the hardness caused by making the metal for their spring-like quality, and not upon any tempering. It is this kind of a spring that we find in nearly every Bourdon gauge, so why should we wonder at their failures?

When a steel spring is tempered, the application of sufficient heat relieves the metal which composes it of all internal strains,

besides imparting to it the quality which causes it to recover its normal position and shape after its load is removed.

Mr. Bristol has overcome this multi-strained trouble in certain springs used in his gauge, by forming a delicate spring of electro-deposited metal (I believe he uses nickel). In order to do this, he forms an irregular zigzag circle of some soft material like wax, and then places that in his bath or solution and deposits the required amount of nickel over it, and, after the spring is thus formed, the wax is melted out.

Of course, the metal thus deposited is free from all internal strains, and I see no reason why a spring thus made would not work well for a long time, with the light strains put upon it in its regular service, and I certainly believe it a much better spring than those of hard-drawn brass, found in most of the Bourdon gauges, though, of course, no metal can equal tempered steel for a spring.

Mr. Thomas R. Almond.—I think Mr. Cary is getting very close to the root of the subject. After we subject metal to the distorting action of the hammer or the rollers, or whatever it may be, we get the molecules into an abnormal condition. They require time, vibration, differences of temperature, to get them back into a normal condition. Where we draw brass tubes we have certainly upset the metal and put every molecule into a condition where it has a desire, as it seems to me, to get back or to get away from that condition, and until it gets away from that condition it will be constantly making the effort to do so.

A very interesting thing has come to me lately in regard to the coiling of steel wire. I had a large quantity of it to coil, and it was so hard that I was unable to coil it. It would continually break, and in despair I put it to one side, supposing that it was to be quite a serious loss to me. But the idea occurred to me that if I ran it through a bath of boiling water as it went into the coiling machine I might succeed. I rigged an apparatus so that it could be done in that way, and every bit of the wire has been so coiled without a single breakage, whereas, before that the breaks would occur every few inches. They would occur so often that I could not get a suitable length. It was necessary that I should coil the lengths into 16 feet. Mr. Cary has said that the tempering process will bring steel into a condition where there is not an abnormal tension. I cannot quite agree with him, because this experiment with the water proves to me that there is a continual tension of the mole-

cules, that they are all under strain, and they are ready to take a new form as soon as the temperature is raised to a sufficient degree to permit them to do so. This has shown itself in this practical way; that is, that I have been able to change the form of the steel, and change it so satisfactorily, when it could not be done before, and because it has all been done by change of temperature not amounting to more than 152°. It would seem to me to be very important to consider this side of the subject, because it will not only relate to tempering steel, but to every piece of metal we are handling. I do not consider that this relates to a metal such as Mr. Cary speaks of in the electric process of disposition. I should think that that would be the condition in which the strains are the least; in such a case I think the molecules will be arranged in as nearly a normal condition as possible. Of course, when you take such a metal as that and manipulate it by hammering or rolling, or any process—spinning if you like—that must put the particles into a condition in which they do not like to be. We are crowding them out from the condition which we may assume to be the natural one, and Mr. Holloway, I think, has started a point which is very well worth considering in this connection. Perhaps it is of more importance than we may want to give it, or generally care to give it. The idea that the Bourdon tube when it is new will give us certain results is because at that time we determine what the results will be; that the same results will be obtained from such a tube a year later, is not a fair deduction to make. It is absolutely wrong. I do not think there is any instrument that we may make practical use of that will not change, and change continually.

After a metal has been manipulated, the first crystals formed are very minute. There is a gradual tendency for them to become larger. The tendency is accelerated by variations of temperature and vibrations. These, constantly acting, will develop larger crystals. The crystals will go on increasing in size until the adhesion becomes less, and we get those mysterious breakages which constantly occur. This subject is, to my mind, one of the most interesting subjects that we, as mechanical engineers, should consider.

Mr. G. C. Henning—The question which I asked was not with a view of criticising Professor Bristol's gauges, which are vastly different from the Bourdon, which we know is a variable machine. The difference between Professor Bristol's gauges and all the

others is that he uses the mechanism with infinitesimal motions, while the Bourdon tube acts with considerable motion. In the Bourdon tube the metal is as thin as it possibly can be with the comparatively short tube and a great deal of motion which is largely magnified by mechanism. If a tube varies ever so slightly the variation is magnified many times by the mechanism. We have the Bourdon tube like this (Fig. 329) connected to the multiplying mechanism at *b*, and this tube must be very fine in order to respond to the changes of pressure within. Now, Professor Bristol does the other method. He takes a very strong tube, and in order to make it flexible gives it great length, and solders at the points *a* and *b* (Fig. 330) a strip of flexible material—flexible in the direction of motion, but strong enough to return like a spring. Now, his multiplication is obtained, not by any mechanism as at *b*, *c*, *d* (Fig. 329), which might get out of order, or which might have back-

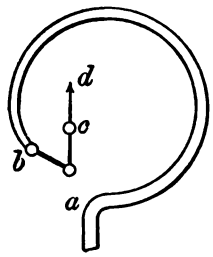

FIG. 329.

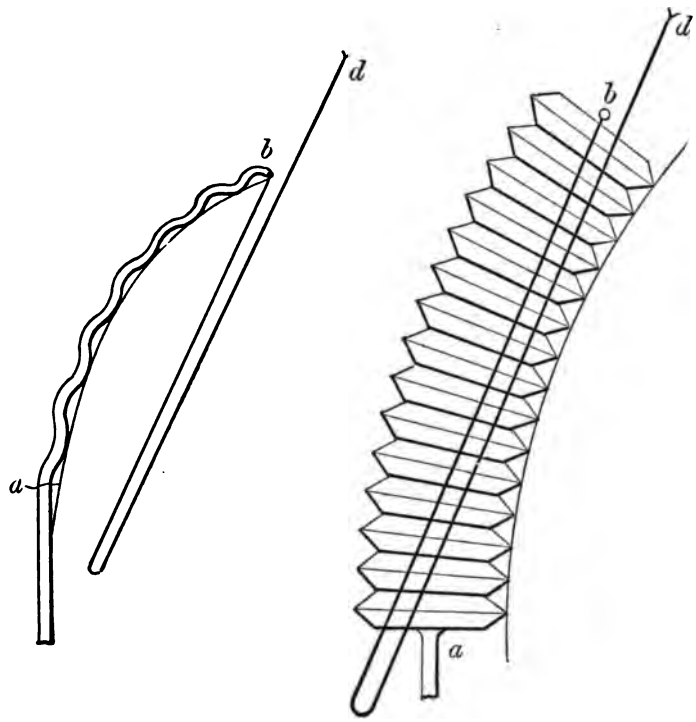

FIG. 330.

lash, but simply connecting from the movable point *b* to a point further distant, at *d* (Fig. 330), and then let a pen *d* write a line. He uses different tubes for different pressures. The result is this, that with these connections very slight changes of pressure will bend this strip, *a b*, of metal in a direction to the right, and as that is a mere strip of sheet metal, great flexibility is obtained which does not bend the strip permanently, and the tube, not being expanded on account of its great resistance, will always return to its original condition. In the Bourdon tube, just as soon as you overstrain it, it will never return, while the Bristol tube does return to its original position so closely that no variation has as yet been seen. In the new form there is a spring of one inch in diameter—a Bourdon tube—but that one inch diameter gives him a length of spring fourteen inches, because he takes four revolutions of the spring. Then again he has no mechanism between its free end and the writing pen which will vitiate the results of the spring, because he connects to the end of that tube at *b* (Fig. 331) a long lever, and the only resistance to the motion is the friction between the pen *d* and the chart. As the paper is moved by clockwork, independent of the tube or the pencil arm, it makes no difference how that clock is affected; the reading will be correct. But the question that I asked was, whether the slight expansions of that tube will not gradually change the position of the free end very slightly. I know that the readings of the same pressures will always be at the same interval because the tube will not change, being heavy in comparison to the original Bourdon tube, which is very light; but I thought the initial position of the point of the writing pen might change on account of some slight variation in the tube. I know it is not the Bourdon tube in the original sense, and is far superior to any other gauge that has ever been made in the way of a recording gauge. Mr. Holloway said that it was difficult to read on a diagram on which this indicator runs on the radii. There is no difficulty at all. Instead of turning a dial, that clock may revolve a cylinder under the point. It is simply a matter of design, and it is most convenient to put a gauge up against the

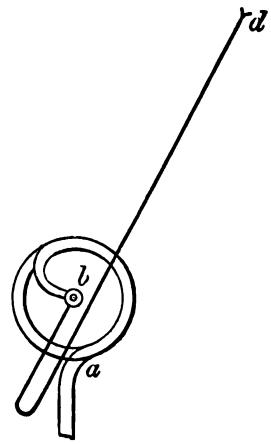

FIG. 331.

wall plate or any place, instead of having a revolving cylinder, which will revolve horizontally or vertically, driven by the clockwork the same way, in which case the pen would write a line on the moving paper on the drum. It simply requires that you have a continuous paper. The way Professor Bristol uses it, you can leave the disk of paper in place as long as the clockwork runs, and the curves will simply overlap each other. In other gauges, like the Edson gauge, you have to remove your paper when the paper is used up, and if that should happen to be at midnight you will not know what happened afterward.

Mr. Holloway.—I simply want to say that what I said in regard to the friction on the diagram, was with reference to what was shown in the paper, not as to what Mr. Henning might invent himself. I believe there is no cylinder shown by which a parallel diagram may be made. I dare say Mr. Henning could very readily make one. Mr. Henning made another point which I cannot quite understand. I always thought it was easier to stop a wheel by getting hold of the outside rim of it than by catching it near the hub; so I think the friction on the outside of the periphery must have a greater effect on the action of the mechanism than if produced near its centre of motion.

Mr. Henning.—The radius at which the friction is applied against the spring is not changed. The friction is always applied at the same distance from the centre of force.

Mr. Holloway.—I mean the friction of the mechanism that turns the disk.

Mr. Henning.—That has nothing to do with the accuracy of the pressure-readings. Friction would affect, perhaps, the time-readings. If you hold your wheel at the edge, of course that would stop it. But the friction to the gauge-spring is always applied just so far from the centre of motion.

Mr. Ball.—Mr. Henning has spoken about the friction of the pencil or whatever makes the record on the paper. As I understand it, in the gauges I have seen, this record is made with ink, and the ink flows from a pen that does not touch the paper. It does not seem to me that that friction could be measured.

Mr. Henning.—I wish to say that the liquid used in that pen is glycerine and water and aniline color. That liquid friction between paper and gauge, on a VERY sensitive gauge, is an appreciable thing, but it is always constant. I did not mean to say that it was a pencil, any more than it simply draws a line.

It is simply an open tube in which there is a solution of aniline color in glycerine mixed with a little alcohol or water. Unless the liquid soaks the paper and the pen sticks in the paper there is very slight friction, and we can neglect it.

Mr. Holloway.—I do not want to take up the time of the Society, but I think it is due to the members that I should say that I did not know the thing was greased.

Mr. James McBride.—I think Mr. Holloway's point is well taken. While the Thompson siphon recording device is a good one, it is troublesome to attend to, and the engineer or attendant is likely to substitute a pencil in its place. This creates friction on the outside of the diagram and gives a wrong record. I think I have seen some of them fixed in this way.

The President.—Do you think it is worth while putting up instruments where the gentleman in charge would allow the men to do such things?

Mr. McBride.—No, sir; but they do it.

The President.—Then there is no use in putting them up or designing them?

Mr. Holloway.—Mr. Bristol puts a lock on them.

Mr. Cary.—There is one point touched upon by Mr. Holloway of no little interest, and that is the effect of temperature upon a spring.

I have tried many experiments to answer this question of the effect of temperature upon a spring (experimenting with tempered steel springs), which have led me to the conclusion that, as long as the temperature to which the spring is subjected when in actual use does not exceed the temperature to which the metal is raised in the final heating of the tempering process, the spring will not be affected in the least.

To make clear what I have just said, you all, of course, know that in tempering a steel spring, it is first heated to about a cherry-red and then is plunged into a cold bath and cooled. This leaves the spring, in most cases, too hard and brittle, and, in order to reduce this hardness somewhat, we have to heat it a second time, but to a much lower degree, and this heating produces, on a polished part of the metal, what are known as the temper colors.

Finally, when the required color is reached (due to the rise in temperature in the metal) a further increase of heat is checked by a sudden immersion in the bath, and the spring is drawn (as is commonly said) down to the required degree of hardness. This

second and lesser process of heating is the one I refer to, and if this temperature is not exceeded, when the spring is in use, temperature will have no effect upon its lasting qualities.

Mr. Almond has mentioned one point with which I am afraid I cannot agree, and I feel sure that it is merely an error in observation, which further experimenting will correct.

He speaks of passing his tempered steel wire through a bath of boiling water, and claims that this temperature (which, of course, cannot exceed 212°) affects the temper of the metal, reducing it from a point of temper too high to coil on a small mandrel down to a temper low enough to do this coiling successfully.

I was once connected with and interested in a long series of spring-tempering experiments, which were made to decide points in question during the course of a patent suit that dragged through the courts for a number of years. This very question of the effect of low temperature on steel springs was most prominent in the suit.

This suit was carried from one court to another, and the evidence that a temperature below 300° would affect the temper of steel was proved so conclusively to be a fallacy that the suit was decided again and again in favor of the concern with which I was connected, who based their claims on this point.

These experiments proved that the temper of steel was affected in a temperature above 300°, and all the temper colors were produced between that and 700°; or, in other words, you can reduce the hardness in a piece of steel that has been made as hard as possible, by the first regular operation of tempering, down to any of the degrees of hardness represented by the different temper colors, by merely putting it into a bath (air or otherwise) heated from 700° down to a temperature a little over 300°, according to the degree of hardness required.

Mr. C. W. Baker.—It should be remembered that the card is revolved by clockwork, and unless the friction is great enough to interfere with the action of the escapement an increase or decrease of friction would have no effect whatever in interfering with the correct record. Even if the friction were great enough to slightly retard the revolution, the amount of pressure recorded would be absolutely correct. The only error would be that the time at which a given pressure was attained, as shown by the card, might be a few seconds earlier than the actual time.

Prof. J. Burkitt Webb.—In the discussion on springs Mr.

Almond stated that he had softened hard steel wire by boiling water, and lest this should be supposed to be at variance with the wisdom of Solomon, that there is nothing new under the sun, I will read a passage from *Bourne's Dictionary of Mechanics and Engineering*,* published by Appleton in 1851. The paragraph is as follows: "The knife edges for Captain Kater's experimental pendulum were very carefully hardened and tempered in a bath heated to 430°; being then found too soft they were rehardened and tempered at only the heat of boiling water, after which they were considered admirably suited to their purpose."

Now Captain Kater's experiment is described in the *Philosophical Transactions* for 1818, so that the use of a boiling water temperature to soften steel is not new, though Mr. Almond may use it in a somewhat different way. The knife edges of Captain Kater's pendulum had to be especially hard, and there is an interesting circumstance connected with these knife edges. Their distance apart had to be measured with great precision by means of a microscope, and it was found that there was a difference of six ten-thousandths of an inch between the results according to whether the background used was white or black.

Mr. J. F. Holloway.—I did not get the impression that Professor Webb seems to have in regard to Mr. Almond's remarks. I thought he said that the use of hot water was for the purpose of preventing steel springs from breaking more. He found that in drawing them through hot water the results were better.

Professor Webb.—The wire was so hard that it broke repeatedly in the process of winding the springs; the hot water softened it and prevented the breaking. That is the interpretation I put upon what he said.

Mr. Holloway.—In the paragraph you read, didn't you say the knife edges were hardened?

Professor Webb.—They were hardened in the usual way, by heating red hot and suddenly quenching them; this made them too hard, and so the temper was drawn by immersion in a bath at 430° Fahr. This made them too soft, so that they had to be hardened again, after which the temper was again drawn, but only as much as a bath of hot water would do it.

Mr. Almond.—The gentleman misunderstood me. I had bet-

* In subsequent editions the name of Bourne is omitted, and the book is known as *Appleton's Dictionary of Mechanics*. In the latest revised edition of 1880, called *Appletcaon's Cyclopædia of Mechanics*, the paragraph does not appear.

ter state exactly what the material is that I referred to. It is No. 11 tempered steel wire furnished by the Washburn & Moen Company, and the request was made in the order, for a wire so hard that it would just bend over a mandrel, 270 thousandths of an inch in diameter. To get that, and have it so hard that it would answer my requirement and yet not break, is a very difficult thing to do, and to return it to them, if it is too hard, brings about the condition that it may be too soft when it comes back. That is to say, thinking it is too hard they get it softer than we really want it. For my purpose I want the wire as hard as possible so as to get a very strong spring from the wire. As you can understand, .270 is a very small diameter. Now this wire is already hardened and tempered. I take it that if you warm a metal that has been compressed by hammering, drawing, etc., you help the molecules to arrange themselves, or bring themselves nearer to what we may call the normal condition, and whenever you do that you will get a stronger material. Now I have proved, by having coiled up 500 lbs. at least of very hard—fire-hardened steel—wire, that by passing it through water at 212° you can form a continuous coil without breakage. I have coiled one piece twenty-five feet long, and, as a matter of fact, a boy sixteen years of age has run a full 500 lbs. of this wire—that had been condemned because of being too hard—through the machine, by its having passed, during the process of coiling, through water at 212°. The whole of the coil has been coiled successfully without any breakage, where breakages would occur previous to that every six, eight, ten, or twelve inches, or something of that kind.

Mr. Holloway.—Was the temper drawn by the fact of passing it through water of that temperature?

Mr. Almond.—That would depend on what you mean by temper drawn.

Mr. Holloway.—To an extent to be appreciable.

Mr. Almond.—Not to be appreciable. I do not think that the temper is drawn, in relation to the value of the material for cutting purposes; but it is as far as strengthening the material is concerned. When I say temper I mean drawing it so that it reaches some particular color with which we are familiar. I think that, whenever you heat a metal and also subject it to some vibratory action, you permit the molecules to arrange themselves according to a law, which undoubtedly exists, and by such an arrangement bring them into a condition which gives much

greater strength to the material, and I think this will relate to any of the metals within certain limits. We all know what a disturbed condition the molecules must be in after hammering, rolling, drawing, spinning, or any of the processes that metals are subjected to. As Mr. Brashear said to me yesterday, it is not a large number of degrees that we are considering; we are often considering a fraction of one degree. He cited the case of rock salt, which is a very peculiar substance, and which has been lying, as you may say, dormant for centuries. The molecules have got into what I consider an absolutely normal condition, and a substance under such circumstances will show many different qualities from what it will show under circumstances such as that of bringing matter into shape for the practical purpose for which we use it.

Professor Webb.—There are some points Mr. Almond has not made quite clear. He stated that the wire was run through boiling water. Now, did it remain hot while it was being wound? He said also that this wire was tempered steel wire, made to order. Now, was it tempered in the ordinary way by heating, quenching, and drawing the temper? I should like to hear exactly the method of treatment and whether the wire showed a color or was bright afterwards; that is, whether there was any color evidence on the wire itself to indicate its degree of hardness.

Mr. Almond.—I am speaking about a fire-tempered wire, *i. e.*, wire passed through a furnace and cooled from a red heat in a suitable medium. Many different concerns have different ways of cooling their wire. Many manufacturers of wire draw or temper their wire by different processes. They all draw it. That is to say, they get it out of the hardened condition which it was in after being made red hot and cooled.

Mr. F. A. Halsey.—Another illustration of the influence of comparatively low temperature upon steel is found in the operation of straightening files by the file makers. The files are heated in a lead bath and quenched in brine, from which they are removed while still warm, and in this condition are straightened. I have no definite idea of the temperature at which the straightening is done, but it will be sufficient to say that while the files are warm enough to give off a slight cloud of steam, they are cool enough to be handled by the bare hand. In this condition they are straightened with surprising facility and dexterity, although

if allowed to become stone cold, they would snap like pipe stems under the same treatment.

Mr. George R. Stetson.—In order that the younger members may not feel that we are dealing in exact science, I would say that I was led to a series of experiments at one time to break bars of round steel about an inch or an inch and a quarter in diameter. This work I desired to do under an ordinary Fowler or drawing press. I was led to the experiment by hearing that the steel jaws used in facing bench vises which might be $\frac{5}{8}$ of an inch thick by $1\frac{1}{4}$ inch face were broken by putting them in hot water and cutting them into lengths under a press.

I desired just the reverse of the action which the hot water had in the case of the springs, and expected the steel to break much more easily. I fixed up such a bath and tried the experiment. I had large quantities of steel to get to a certain length. I was not fully satisfied that it was a fact that it did break more easily, and abandoned the experiment without practical results; so that, by the use of hot water, I desired to produce exactly the opposite results from Mr. Almond.

As to the matter of temper in hot water, in the instrument that was mentioned by the Professor, I think that you would have to study the quality of the stock to produce the desired results. In very high carbonized steel, in the shock from the low red heat (which it is desirable to give to that quality of steel) to the immersing in entirely cold water or brine, you are very liable to produce cracks; and it is very common among practical blacksmiths, if that condition prevails, to warm the water. By the higher quality of the steel you get a solid piece of work, and it may be very hard, but under common conditions a blacksmith would never think of warming the water to produce the hardest results, and our common practice of drawing the temper of springs by burning the oil shows where the heat has to be carried to get a spring under ordinary conditions. Generally the flashing point of a common animal oil is the point at which the spring is immersed, and if you want a kind and tough spring you will immerse to cool in oil, not in water. If you want it harder, or a lower grade of stock were used, you would probably put it into water more or less hot. The effect produced would depend on the character of the stock.

In the discussion I wished to have it brought out whether a steel spring would be affected by a heat not greater than an ordi-

nary boiler pressure of from 100 to 120 lbs. We would like to know whether a spring tempered at 600° of heat would be deteriorated by a heat of from 212° to 230°, and whether, in the spring that was being described, this heat would have a deteriorating effect. I think there is a constant motion in the molecules of steel, and it does change in time; and with moderate heat it would probably change more rapidly than if the heat were less, or the more regular the temperature or the lower the temperature in which the spring worked, the longer it would be liable to live and retain its elasticity.

I am satisfied of the correctness of the theory brought out in the discussion as regards the springs and the gauges, that the spring that has the least work to do will be the spring that will carry its gauge the longest and the most surely. I believe in the fatigue of metal. I had some experience in running an electric railroad. We had an engine in which the governor was of the centrifugal weighted and spring type. We got the best springs possible, but the railway service kept that spring so active that it would last but a few weeks; whereas, if it had been on a regular load it would probably have lasted a good many years. It was worn out by its constant motion trying to govern the engine, and I am satisfied that that takes place wherever a spring is constantly active.

Mr. Cary.—I would like to ask Mr. Almond how long it was, after he first tried to coil that wire into springs, before he coiled it successfully, using the hot water. As I understood him, he said he received this wire and was obliged to lay it aside, thinking it was useless.

Afterwards he took it up, using this process he described, and succeeded in coiling it. How long after his first trial was it coiled successfully?

Mr. Almond.—I shall have to state this case precisely as it occurred. The whole subject was new to me until two or three years ago. The first wire I got for the purpose was from Cary & Moen. Mr. Cary, who is present, was once with that concern, I believe. The wire was not satisfactory. It would keep breaking, and it was one of the greatest disappointments to me to find that I could not get a wire that would not break. They all told me that my requirements were too severe, that I never would succeed in getting a satisfactory material. I finally went to the firm of Miller & Van Winkle of Brooklyn. They are within half a dozen

blocks of my place, and they were kind enough to do what they could for me in the way of tempering the wire. They said, "Mr. Almond, if our wire is too soft for you we will make it a little harder, and we will keep on making it harder until it is so hard that you cannot use it." Well, they did. The hardest wire they gave me was a little short of what I wanted. Now, I found that in repeating my orders I did not get the same results, and, as a matter of fact, I never have succeeded in getting such results as would give me uniform conditions. In despair I went to the Washburn & Moen people and asked them what they could do, and they said they did not know what they could do, but they would do the best they could. I gave them an order for 100 lbs. I am talking about No. 11 tempered steel wire as manufactured by the Washburn & Moen people. Everybody who knows anything about tempered steel wire knows that it is not a hard-drawn wire such as suggested by Mr. Webb. It is a fire-tempered wire; that is to say, a wire that has been made red hot and cooled in some suitable medium, be that what it may. Every manufacturer has a different one, I believe. Some have patented processes for that purpose. They gave me 100 lbs. of wire that was coiled in a machine of my own invention, which is being patented now. Every bit of the wire went through the machine without any break, and the strength of the spring was so good that it seemed to me I had at last solved the problem of getting a wire that suited my purpose. After this was coiled up I thought, the best thing I can do is to send for the representative of the Washburn & Moen people and tell him to write out an order of such a character that I could use it for all time in ordering just that kind of wire. He sat down and wrote the order out, and he took it with him. I kept a copy of his order, and they sent me another hundred pounds. I started to coil this, and coiled perhaps six inches, and it broke; then twelve inches, and it broke; then perhaps two feet, and it would break, and it kept on breaking in that way. I could not coil more than from six inches to two feet at the most.

A Member.—Of wire or spring?

Mr. Almond.—I mean that the spring would break or the wire would break after six inches of spring was coiled. When I say coiling the wire, I mean producing the spring. Now there was 500 pounds from Miller & Van Winkle of Brooklyn that I could not coil. It was too hard. It had been there perhaps five weeks.

It had been bought since the discussion of this subject at our New York meeting. It had not been in my place for a long time. I had not put it aside to season, as I expressed myself at our last meeting when this subject was discussed. Now, you see, under the circumstances I was compelled either to lose the wire, abandon the manufacture of flexible metallic tubing for practical purposes, or find a means by which I could get the wire into a condition suitable for the purpose. I do not think that ever any man's nerves have been put into such a distressful condition as my nerves were by this subject. The idea of standing there with your shirt sleeves rolled up, day after day, spending several hours attempting to do something which it seemed impossible to do. For three years I have been working on this subject, and it is not until within three weeks that I have discovered a solution of a problem that seemed to me something that possibly I never would be able to get around. I have got around it by simply passing that steel wire through a bath of hot or boiling water, which is situated within twelve inches of the machine into which the wire is passing as it is being formed into the spring. Now I know that the wire will be nearly 212° in the bath. I know that the temperature will be reduced somewhat as it passes into the machine. I do not think that it will lose more than a few degrees, say 10°, 12°, 20° if you like. But I am so convinced of the value of that small increase of temperature in rearranging the molecules, or permitting the molecules to rearrange themselves into a condition further removed from that of tension into which they get after they have been submitted to such a disturbing action as red-hot heat or to such a disturbing action as the drawing of wire, hammering the metal, or anything of that kind, that I thoroughly believe that, if we consider the subject fairly without reference to anything such as Mr. Stetson has described of the charring of a piece of wood, if we look at it fairly, we shall get to understand that we can put our metals into a much better condition by looking at the molecules as they are in the mass, and if they are in a condition of tension get them out of that condition of tension, and so improve the strength of the material. I hope I have made myself clear.

Mr. Cary.—The reason I asked the question concerning the length of time between the two trials of coiling this wire into springs was, there seems to be a very peculiar action going on among the molecules of steel for some time, directly after it is

tempered, and this is much more noticeable in highly tempered steel and in steel containing much carbon.

At one time I had charge of a large manufactory which turned out many tons of springs every week, and also made the steel wire as well, from which these springs were made, and thus I had an exceptional opportunity to observe the many peculiarities of both the springs and the wire, and certainly nothing was more puzzling than these changes which took place in newly-tempered steel.

Often tempered wire was sent to the spring department to be coiled into springs, and found too hard to wind on a mandrel without breaking, and often this wire would be put one side for several weeks, after which time it would be wound on the same mandrel without difficulty and without breaking. It may be that this may explain the success Mr. Almond had in finally coiling springs successfully after failing in his first attempts.

Those who have made much use of thermometers know that a change often occurs in them with age, which is also probably due to a molecular change caused by the glass being under internal strains (similar to those in tempered steel) when it leaves the hands of the maker.

Nearly all, if not all, thermometer makers who lay any claim to making instruments of precision keep their glass tubes, after receiving them from the glass works, for many months before using them, that the molecules composing these tubes may cease this strange motion among themselves, and even then, after blowing the bulbs, filling the tubes with mercury and sealing them, when an instrument of very great precision is required, we find them again laying these aside for some time before finally marking them.

Thus, these peculiar molecular changes do not seem to be confined to tempered steel alone.

Few people, outside of those in the steel business, can appreciate the great dissimilarity found in different steels. Not only is a very marked difference found in the steels made by the various processes, but in various samples of steel made by the same process; and analyzing the same we find such marked dissimilarity that it is necessary to treat each one quite differently when working it, such as in the manufacture of wire.

Certain steels are adapted to certain uses and are unsuitable for others. To illustrate, steel wire used for springs would

be useless for making fish-hooks; fish-hook wire would hardly be adapted to make umbrella ribs, while wire adapted for umbrella ribs would not be found the best for making cutting instruments; and so I might extend the list indefinitely.

When connected with Cary & Moen, in New York, the concern referred to by Mr. Almond, we were, for the reasons I have just explained, obliged to carry several hundred kinds of steel in stock to meet all the demands of customers.

People often came into our offices and ordered simply "steel wire," and when we would ask them what use they intended making of it (in order that we might judge which kind of steel was best adapted to their purpose) they would often give us no satisfaction in their answers, thinking we were trying to discover their secrets, and the result would be, they would receive wire made of a stock as nearly right as could be guessed at, which, unfortunately, would not always be adapted to their purpose, and, of course, the manufacturer was blamed for the fault of the purchaser.

The question was asked, a moment ago, how wire was tempered.

By referring to the sketch I have made (see Fig. 332) you will obtain somewhat of an idea of what is known as the "Waterman continuous process," which is the method used by all manufacturers in this country for tempering steel wire.

I have shown everything in section, and thus the dotted line, running from one end to the other, represents the wire being tempered.

The wire is first wound, in considerable lengths, on reels, and to accomplish this often short lengths are spliced together. One of these reels is placed on a fixed arbor, as shown at *R* 1.

From there the wire passes through a bath of molten metal contained in what is called "a tempering pot" (*T P*), beneath which is a fire, generally of coke, which keeps the molten metal at a constant temperature.

The shape of this tempering pot is shown at *A*, and is like the letter D. It has one small opening at each end on the top, into which the wire passes at *B*, and another corresponding opening at *C*, out of which the wire passes, heated red hot.

By using this large metal bath a great uniformity of heat is secured, and by reducing the openings at each end to a minimum size, the liquid has but little contact with the air, which reduces its oxidation to the smallest possible amount.

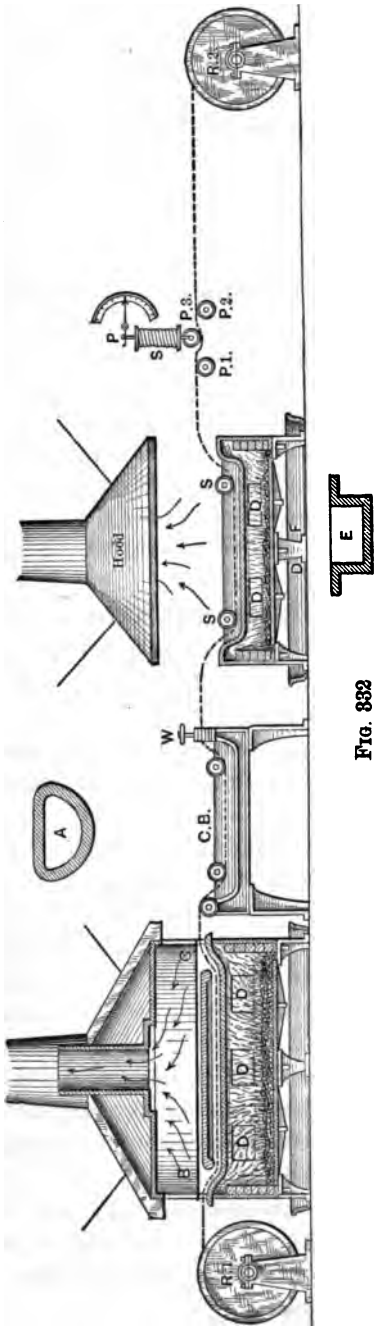

FIG. 332

After leaving the tempering pot at *C* at the required temperature, the wire is plunged, as it moves along, in the cooling bath *CB*, which consists of a galvanized iron tank filled with oil, and this tank is placed, as shown, into a large tank filled with water. Both oil and water are made to run rapidly in and out of their respective tanks through pipes (not shown), and thus the oil is always kept cool.

We have now completed the first operation of tempering our steel wire, that is, the hardening—and the wire is about as hard and brittle as it can be made—and next we must "draw the temper" to the required degree of hardness, which is accomplished by passing the wire through an open lead bath in the drawing furnace (*EF*). Just after the wire emerges from the oil bath the surplus oil is wiped, by wipers placed at *W*, which prevents the oil drippings from being carried into the drawing bath. It is at this last furnace that the greatest skill is required to produce the desired temper in the wire. Small grooved pulleys cause the wire to be submerged in the lead bath. By manipulating these pulleys, the wire may be made to pass through the entire length of this bath, or else it may (by using one pulley) be made to just barely touch the surface of the lead at one point.

This allows quite a difference

in the time the wire is exposed to the heat, and still a further regulation of time of exposure is obtained by altering the rate of speed by which the wire moves. This is done entirely by altering the speed of the receiving reel (*R* 2), which is accomplished by means of speed cones, and the entire power which moves the wire is delivered through this last reel.

The shape of the open lead bath tank in the drawing furnace is shown in section at *E*.

The oil generally used in tempering steel wire is sperm oil, which, however, is sometimes mixed with common fish oil.

This is practically the whole of the Waterman process, such as used by Cary & Moen, Washburn & Moen, and all other wire temperers in this country.

Of course, by this process one is not able to work to the temper colors, as is done by blacksmiths and others, to guide them in drawing the hardened steel to the required temper. A long experience is necessary for the successful operation of the Waterman process.

The lack of anything visible to guide the workman in his work, during this operation, led me to invent a simple instrument, which I placed between the drawing furnace and the receiving reel. It consisted of three pulleys, with grooves to receive the wire. Two of these were placed a definite distance apart under the wire, as at *P* 1 and *P* 2 (see Fig. 332). The third touched the wire on top, midway between the others, as at *P* 3. This last pulley was pressed down with a definite force by means of a compression spring (*S*), which bent the wire at this point.

The bending, of course, was greater when the wire was soft, and less when it was harder, and this difference of deflection was indicated by the movement of a pointer (*P*).

This, I believe, is the only instrument ever made for indicating the degree of temper in steel.

The kind of steel best adapted for springs seems to be an open hearth steel, comparatively low in carbon.

Many people select a steel high in carbon for making springs, but in doing this they make a great mistake, as the higher the percentage of carbon, the more difficult the process of tempering becomes, and the chances are the spring will be spoiled in this operation.

With a steel low in carbon, the process of tempering becomes comparatively easy, and a great uniformity will be found to exist

among a large number of springs, after they are finished, and breakage, while in use, will be found to be reduced to a minimum. If this comparatively low carbon steel is properly selected, the spring made from it will be found as good, durable, and lasting as the one made from the very highest grade of steel.

As to Professor Webb's remarks about certain knife-edges being tempered a second time, and the hardness drawn in boiling water, after these same knife-edges were originally tempered and found too hard, I believe that this is easily explained; and I cannot believe that the boiling water affected their temper in the least; in fact, I believe that after the second hardening and before these pieces were put into the boiling water, had they been tried in their place, they would have proved to be all right, the same as they finally did.

I have often found in actual experience that I could never harden a piece of steel as hard the second time as I could the first, and for that reason, in the steel wire business, we would always avoid re-tempering wire if it were possible to do so.

I think that this will fully explain the result obtained in the case quoted by Professor Webb.

*Mr. W. H. Bristol.**—In designing the different recording gauges that I have had the honor of presenting to this Society at different times, the object has been to produce forms which would be fundamentally simple in construction, and entirely free from the usual multiplying mechanism which is liable to fail.

These instruments are the result of a large number of experiments, and have been finally decided upon with confidence that they would remain constant and give the most satisfactory results in continual use.

Our expectations have been fully realized in the case of the sinuous and diaphragm forms of gauges, and as we find in our experiments with this new form of helical tube that it will stand even more severe tests than the sinuous form of tube. I have felt very safe in concluding that it would remain constant during years of service.

This helical form of tube has only been in use for a few months, but as far as experience goes they have given the most satisfactory results.

As to the effect of heat upon the gauge, would say that I have never made any experiments, but believe it cannot help being

* Author's Closure, under the Rules.

detrimental to the gauge. Every steam-gauge should be provided with a syphon to prevent the steam reaching it.

The recording pen touches lightly with a constant pressure, whether near the centre or circumference. The friction caused by the pen is very slight, and the difference in the resistance offered to revolving the chart, whether pen is at centre or circumference, is so small that it would have no effect on the strong clock movement which is provided for revolving the chart.

DCIII.*

COST OF AN INDICATED HORSE-POWER.

BY DE COURCY MAY, NIAGARA FALLS, N. Y.
(Member of the Society.)

I DO not believe there is any subject in connection with the steam-engine on which there is such a diversity of opinion as the cost of an indicated horse-power per annum.

Much has been written about the various types of engines, and the advantages of each, of cylinder condensation, point of cut-off, etc., and many and most elaborate tests have been made to gather more or less valuable information concerning the steam-engine and its performance. I find, however, very little reliable data as to the cost of operating steam plants, due partly to the care with which the cost of anything is concealed, but more probably to ignorance on the part of the owners, who in most cases have only vague ideas as to the actual cost of their power. Two papers on this subject have been presented to this Society—one by Mr. Manning, on the "Comparative Cost of Steam and Water Power;"† and one by Mr. Main, on the "Cost of Steam and Water Power."‡ In the engineering papers, paragraphs have appeared from time to time, purporting to give the cost of an indicated horse-power with more or less accuracy. Our text-books scarcely treat the subject at all. Mr. Manning has given us a very interesting paper and some valuable information about the water-powers of Manchester, Lowell, and Lawrence, and the cost of the machinery for utilizing them. In comparing the cost of water-power with steam, the steam-engine used is, I imagine, an ideal one, and not one taken from actual practice. It is an engine running 10¼ hours a day, and delivering 1,000 actual horse-power, and indicating 1,100, and running, presumably, at full load the whole time. He takes the cost of a

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*Presented at the Montreal meeting, June, 1894, of American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

† *Transactions American Society Mechanical Engineers*, Vol. X., p. 499, No. 332.

‡ *Transactions American Society Mechanical Engineers*, Vol. XI., p. 108, No. 360.

modern compound engine of this size, including piping and foundations, at $27 per horse-power, and the boilers ready for use at $10 per horse-power, the consumption of coal being at the rate of 1¾ lbs. per horse-power per hour. The results of his calculations, after allowing for interest and depreciation on plant, running expenses, repairs, etc., is, for this engine, $22 per indicated horse-power per annum, 309 days of 10¼ hours each. No exhaust steam is used for heating or any other purposes. The cost of the coal is $4.50 per ton of 2,240 lbs.

Mr. Main in his paper also works out the cost, presumably from an ideal engine of 1,000 horse-power, giving very complete figures for the compound, condensing, and high-pressure types; the compound engine, with piping, costing $25 per horse-power, and the total steam plant, including buildings, being put at $58.36 per horse-power, Mr. Manning's figures being $52.50 for the same thing.

The result of Mr. Main's calculation is, that an indicated horse-power for the compound engine costs $24.08 for 308 days of 10¼ hours each, no exhaust steam being used for heating or other purposes, and the price of the coal being $5 for 2,240 lbs.

For the purpose of comparing the above figures with those given by other engines, I will give the cost of a horse-power in both cases for 365 days of 24 hours each, keeping the interest, depreciation, taxes, etc., unchanged, increasing the cost of attendance and supplies and coal in the proper proportion, and making the price of the latter $5 per ton in both cases.

This would make a horse-power cost for 365 days of 24 hours each:

According to Mr. Manning	$50.99
Mr. Main	51.75

I have many other papers on this subject, two of them by eminent members of this Society, and all based on an ideal engine working continuously at full load, a condition which is rarely attained, except in a few pumping engines; the cost of an indicated horse-power for 365 days of 24 hours each being much below anything I have been able to get from actual engines working under ordinary conditions.

I think it will be interesting to contrast the figures given above with those given by a very high-class triple expansion

engine working under full load, or nearly so. I will take the Allis pumping engine at Milwaukee, as given in Professor Thurston's paper, read at the last meeting of this Society. The figures given are from a very accurate test, and the result shows this engine to be, I think, without a superior.

During 1892 this engine ran 6,980 hours, pumping 5,204,368,000 gallons of water 160 feet high, and used 5,516,000 lbs. of anthracite coal, costing $5.95 per 2,000 lbs.

The water horse-power from the above figures was 503.33, and if the ratio of water to indicated horse-power had been the same as during the trial, the indicated horse power would have been 554.16; but as the performance was probably not so good, I assume that the average indicated horse-power during 1892 was 560.

The cost of the plant as given by Mr. Benzenberg, the city engineer, was as follows:

Cost of engine (and pumps) and four boilers, including foundations and boiler settings	$81,000
Cost of so much of buildings and boiler house as is chargeable to engine	31,500
Total cost of plant	$112,500

The cost of operating in 1892 6,980 hours with coal at $5.95 per 2,000 lbs. was $28,107.10.

Reduced to 365 days of 24 hours, and coal at $5 per ton (2,240 lbs.), the cost will be:

Coal	$17,275
Engineers' wages (unchanged)	10,680
Repairs to engine and boilers (unchanged)	414
Oil and waste	466
Packing	288
	$29,105
Interest on plant at 5%	5,612
Depreciation at 4%	4,500
Total cost 365 days' (24 hours) operation and maintenance	$39,217

$$\text{Cost per 1 I.H.P.,} = \frac{39,217}{560} = \$70.02$$

This is exclusive of taxes and insurance, and no charge made for the land occupied, and no cost for water; but it is to be remembered that it is a slow-moving machine, and the value of pumps is included.

The number of revolutions of this engine when pumping was about 20 per minute during the test, but was probably slower during the year's run. Speeding this engine up to, say, 50 revolutions per minute, and assuming that it will develop 1,500 H.P., let us deduct the value of the pumps and add sufficient boiler power to increase to 1,500, and base the consumption of coal on that used during 1892, and not on the test; we will then have:

Cost of plant		$112,500
Deduct value of pumps and foundations for same, say	$31,000	
Deduct proportion of building, say	10,000	41,000
		$72,500
Add say six boilers and settings and proportion of house and stack, say		20,000
		$92,500

Say $90,000, which would be at the rate of $60 per horse-power for the complete plant. I have deducted $10,000 from the cost of buildings, as those at Milwaukee are rather monumental in character.

The annual cost would then be:

Coal	$46,215
Engineers' wages (unchanged)	10,680
Repairs (unchanged)	414
Oil and waste 50% added	621
Packing (unchanged, as it included pumps)	288
	$58,218
Interest on plant at 5%	4,500
Depreciation on plant at 4%	3,600
Total annual cost	$66,318

Cost of 1 I.H.P. $\frac{66,318}{1,500}$ = $44.21 for 365 days of 24 hours each, coal $5 per 2,240 pounds.

The consumption of coal per I.H.P. per hour is 1.57 lbs. in the above calculation, and this is, as before stated, based on the operations for the year 1892.

It is stated in Prof. Thurston's paper, that steam was taken from the boilers to drive a small engine used during the day to operate the machine shop, and at night for lighting the building, and some steam was also taken for heating the building. Allowance ought to be made for this steam consumed, but in view of the fact that the lighting and heating are as much a charge against the plant as the oil, waste, and packing used, I

do not think any allowance should be made, except for the power used in the machine shop during the day. The difficulty of ascertaining this amount, and the fact that it is so small in quantity as to scarcely affect the general result, has decided me to make no allowances.

While I do not pretend that the above figures are exact, I believe they are fairly accurate, and rather in favor of the engine, bearing in mind the omission of any charges for taxes and insurance, or any rental or interest on the value of the land occupied by the plant, or charge for water. I think it is safe to assume that one indicated horse-power cannot be produced for 365 days of 24 hours each for less than $40, with coal at $5 per ton of 2,240 pounds.

From such data as I have been able to collect I give in the table below the cost of one indicated horse-power for 365 days of 24 hours each, and 308 days of 10¼ hours each, the 10¼ hours representing the running time in the ordinary mill or factory:

The interest on the plant is placed at.....5 per cent.
The depreciation on the plant is placed at.....5 " "
The insurance on ¾ of the plant is placed at.....1 " "
The taxes on ¾ of the plant are placed at.....1½ " "

No rental or interest on the land occupied by the plant or cost of water is charged.

The value of the land occupied varies so largely, especially with electric lighting and trolley plants in large cities, as to make the insertion of these figures misleading.

	Cost of one I.H.P., 365 days of 24 hours.				Cost of one I.H.P., 308 days of 10¼ hours.			
	Coal per 2,240 lbs.				Coal per 2,240 lbs.			
	$ 2	$ 3	$ 4	$ 5	$ 2	$ 3	$ 4	$ 5
Triple expansion pumping, Allis, 20 revs.	48	55	61	67	31	33	35	37
Triple expansion without pumps, Allis, 50 revs.	27	33	39	45	16	18	20	22
Compound mill, best engine	29	36	44	51	17	19	21	24
" " average	39	46	52	58	22	25	28	30
" electric light, average	122	139	157	174	78	84	90	96
" trolley	48	58	68	79	29	32	36	39
Triple expansion trolley	45	54	64	74	26	29	33	36
Condensing mill	44	52	61	69	25	29	33	38
Non-condensing, 50 to 200 H.P.	70	76	81	88	49	53	57	62

The figures for engines below 50 H.P. vary so widely that I have omitted them.

The above table has been calculated as far as possible from actual engines running under ordinary conditions; but I do not pretend that the figures represent true average values, not having been able to collect information from a sufficient number of cases. The engines are nearly all high class, and of large powers; the figures in the table are, therefore, rather below than above the average.

I hope that this paper may be the means of eliciting fuller and more exact information on what I feel sure is a subject of great interest and importance to the members of our Society.

DISCUSSION.

Mr. Samuel Webber.—I have been much interested in Mr. May's valuable paper, and would like to call attention to the fact that a very interesting paper on this subject was contributed by our fellow-member, Mr. Chas. E. Emery, to the American Institute of Electrical Engineers, last year. That paper I have examined very carefully, and find it to agree very closely with my own observations. It should be noted that while the cost of fuel is the principal item, the *size* of the engine is a very important factor; *i. e.*, that a 1,000 H.P. engine costs much less per horse-power per plant than one of 500 H.P., and the latter bears the same relation to one of 250 H.P., and the same holds good as to running expenses, and as a substantiation of this fact I beg leave to offer the following notes, based on data given me by the treasurer of one of the latest and best mills in Fall River last winter. The data were taken in the months of April, May, and June, 1893, when no steam was used for any other purpose than power, and all care was taken to reduce the cost of that to a minimum. The two engines were at the Globe Yarn Mills, Fall River, both "Wetherell cross compound condensing," and the coal cost $3.70 per ton.

To the data furnished me from the mill I have added charges for sinking fund, interest, and repairs, on the basis adopted by Mr. Manning, which amount to 1% more than those taken by Mr. May, and they give me as follows:

NO. 2 MILL, 1,050 H.P. PER 73 DAYS OF 10 HOURS.

Coal burned, 1,527,000 lbs., cost	$2,519.55
Wages of Labor	555.50
Oil	51.50
Supplies (Waste, etc.)	25.00
	$3,151.55

Equal to	$12.65 per H.P.
Add to this Sinking Fund	3.14 per 5% on cost.
" " " Interest and Taxes	3.77 = 6% on cost.
" " " Repairs	1.57 = 2½% on cost.
Total	$21.13 per Indicated H.P.
Add 10% per Engine	2.11
Gives	$23.24 per Available H.P.

From engine for use in mill and for comparison with water power.

The engine in No. 3 Mill was only 650 H. P., and the figures are as follows for 76 days. Cost of plant, $42,000.

Coal burned, 1,152,380 lbs., cost	$1,901.43
Wages for Labor	463.60
Oil	42.50
Supplies (Waste, etc.)	21.50
	$2,429.03

Equal to	$15.19	per H.P. steam to engine.
Add as before	3.23	" Sinking Fund = 5% on cost.
" " "	3.88	" Interest, Taxes, etc. = 6% on cost.
" " "	1.62	" Repairs = 2½% on cost.
	$23.92	cost of Indicated H.P.
Add	2.39	= 10% for Available Power.

Gives.... $26.31 for results to compare with water power, which is always estimated as delivered from the wheel.

I believe these to be about the minimum figures for mill engines, and that a 300 H.P. plant, such as is very common, would show much higher. My experience has always shown it to be so, with the best single condensing engines. This is a very interesting subject, and I hope other members may give us the results of their trials of the *actual* cost of steam, in their own localities, from engines of different sizes.

Mr. Albert A. Cary.—The cost of a horse-power is a question often put to mechanical engineers by power users and power producers.

How seldom one is able to answer it with accuracy!

If power was produced in all plants with equal economy, it might be possible to formulate some rule which would afford an answer approximately correct for all cases, but as a unit of power costs the producer twice as much and even more in some plants than it does in others, it is impossible to make an estimate of cost without going into all the details relating to the power plant.

In steam plants some of the so-called "horse-power" is furnished as steam direct from the boiler, for heating, for steam-kettles, for vulcanizers, and many other purposes, and then we often find that the steam-power is taken from more than one engine, and one may be the large triple expansion condensing engine, while in some remote corner of the mill we may find another engine, old, patched up, taking steam full length of stroke, and running at odd intervals, and besides these we find the steam-pumps at work taking their 60 lbs. of steam per hour for each horse-power of work they may deliver.

In the face of these and many other difficulties, it is pretty hard to determine by the use of an indicator and many calculations just what a horse-power is, and after deciding this, just what it costs.

There is but one way, that I can see, out of this difficulty, and that is, by taking for a standard horse-power the one recommended by this Society's committee on a standard method of steam-boiler trials (30 lbs. of water evaporated from a feed-water temperature of 100° into steam at 70 lbs. pressure) and then determine, by a boiler test, the cost of producing this amount of steam at the boilers; and then after the steam leaves the boiler, it makes no difference whether it is used in an engine of high or one of low economy, or whether it is used for heating or any other purpose.

Of course, it is not always possible to determine the proportionate part of the total amount of steam leaving the boiler that is used by each lessee, but there are many cases where this can be done, and then it may be said that the expense of running an accurate boiler test made by an experienced engineer may stand in the way of having this done in some plants.

I was at one time connected with a manufacturing concern in New York, using some seven or eight hundred horse-power, and I was obliged to determine this question of the cost of a horse-power.

We had one tenant who occupied the greater part of one of our

buildings, and, besides, we furnished heat and power to one or two neighboring concerns.

The first mentioned concern paid us rental for the portion of the building it occupied, from which figures I could estimate the value of the space occupied by the steam plant, and, in addition, they paid for the steam and power they used at a certain price "per horse-power."

I first ran several boiler tests to determine the amount of fuel required to evaporate the standard amount of water.

Thirty pounds of water evaporated from a feed-water at 100° into a steam at 70 lbs. pressure requires 33,305 thermal units.

My conditions were somewhat different from these, as my feed-water averaged about 187°, and we carried an average working pressure of 80 lbs. A little figuring will show that under these conditions 32½ lbs. of water will be evaporated with 33,305 thermal units, which is, therefore, an equivalent to the standard horse-power, and, therefore, I adopted the evaporation of this amount of water, under these conditions, as my standard horse-power, and did all my figuring on this basis.

After determining the average amount of fuel required to evaporate the unit quantity of water, every other expense which belonged to the "Power Department" was carefully sought out and used in determining the cost of the standard horse-power.

The fuel used was an anthracite egg or broken coal costing about $4.80 per ton (of 2,240 lbs.).

The cost of the engines, boilers, pumps, piping, and all fittings and apparatus about the boiler and engine rooms was taken, and the interest on this investment was included.

The rental which we could obtain for the space occupied by the steam plant was considered, as was also the cost of attendance, oil, waste, packing, and supplies of all kinds, to which was added an amount for depreciation, tax, and insurance.

We tried to include in our figures every item, and maybe went too far in one direction, and that was in calling the power plant one of the departments of our factory (there being some eleven or twelve), and we made it bear its proportionate share in the general running expenses of the business, such as the "non-productive labor," that is, the office expenses, superintendence, salesmen, and all items of that kind.

The result finally obtained was that a horse-power was costing

us about $47, and we charged our tenants, on the average, about $55 a horse-power.

With some little inquiry in New York city, among those leasing power to tenants and neighbors, I found that the price received ran from $45 to $60 a horse-power.

A few years ago I knew of one concern in New York who was selling steam and power at the rate of $42 per horse-power (but I am sure that neither they nor their tenants knew just exactly what that term meant), and I have also heard of a few places in New York city where the charge was over $60.

Of course, it is understood that all of the above-named figures represent the price per year, with ten-hour days, and no running on Sundays or legal holidays. A provision was made for a stoppage of one week in summer, and a second week six months later, during which the steam plant was supposed to be thoroughly overhauled. A further provision was also included in each contract, that, should the plant be stopped through any accident, four additional days were to be allowed in each year to make the necessary repairs. A clause covering strikes and damage by fire or water was also included.

Mr. J. F. Holloway.—This is certainly a very interesting and important subject to engineers in all parts of the country. But it strikes me, from what I have heard, that it is a little like the play of Hamlet with Hamlet left out, and which is, of course, not as interesting as it might otherwise be. The elements otherwise that go to make up the cost of steam-power, must vary very greatly from two or more things which have not been noticed in these papers; one is the type of engine used, and the other is the cost of the fuel by which it is driven, to which is sometimes to be added the cost of the water for the boilers.

The President.—Mr. Holloway, if you will excuse me, add the composition of the coal.

Mr. Holloway.—Also the composition of the coal, and especially as to the quantity of slate in it, and more recently the sizes of the coal. We have had a great change of opinion among engineers as to the money value of the different sizes of coal, and I am glad to know that experiments have been made at a large expense that very likely will change our views as to which is the best kind of coal (as to size) to use. But the point is, in getting at the cost of steam-power, it might be one thing in one city, and another thing in another city, but in order to make a fair comparison we

ought to know the type of engine used, and the cost of fuel attending, and all the outgoing expenses.

The President.—I might say also the cost of getting water for condensing, if you try to do it.

Mr. C. E. Harte.—It may be of some interest to the members of this Society if I were to relate, in some degree, the arrangement of plant which we have recently installed. We have a 600 H.P., cross compound, condensing, Corliss type of engine. The cylinders are 24, 42 × 48.

Our feed-water is passed through a Berryman heater into which the L. P. cylinder exhausts. Then it is passed through a second heater of the same make, into which the feed-pump condenser and automatic-pump exhaust. The cylinders and receiver are jacketed, and the condensation is added to the feed after passing from the second heater. At all these points I have placed thermometer wells so that I am able to get the temperature and determine the value of each of the heating points. We, by this system, get a temperature averaging about 204°.

Our engine is somewhat larger than we need just at present, as we are developing only about 380 H.P., with an expenditure for two hours' work of about 6,000 lbs. of George's Creek coal.

Mr. Cary.—Answering Mr. Holloway's queries, I would say that in figuring the cost of a horse-power, the unit used by me was identical to that adopted by this Society's committee on a standard method of conducting boiler tests as I have already explained.

In connection with this subject, I believe that a description of the methods I introduced and used in the plant I have referred to, will not be out of place here.

I had two tanks built side by side, separated by a brick wall; these were placed below the level of the floor, and all the water passed into them before being fed into the boiler.

On the dividing wall I placed a combination valve, of my own design, operated by two floats, one in each tank, which rose and fell as the water-level varied. This valve was similar in shape to the sketch I have made (which is a top view), the two parallel lines representing the dividing wall which separated the two tanks (see Fig. 333). The water on its way to the tank passed through the pipe marked "water supply" into the valve chamber below, from whence it found but one passage of escape into either one tank or the other according to the position of the valve. A

second pipe leaving the top of the valve (marked "to pump") is the suction pipe, through which the water passed on its way to the boiler, and, of course, it is understood that the water was drawn from but one tank at a time.

The operation of this valve is as follows: water passes into the valve from the water supply and finds an exit through "outlet *A*" and finally fills "tank 1" up to a certain level, when it is automatically closed. While this is being done, the pump (which has a direct connection through its suction pipe with the "inlet *A*" in "tank 2") is drawing its supply of water from that tank (having no connection with "tank 1" that is being filled). During this emptying "float 2" drops with the level of the water, and, as it does so, the spring "*S* 2" is being compressed while the rod *R* is motionless. Finally, when "float 2" drops to the required level,

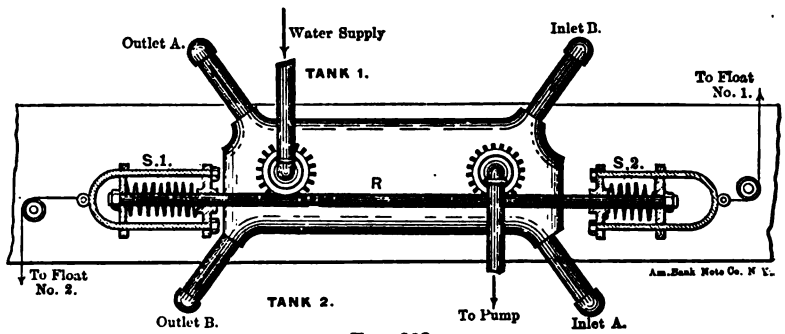

FIG. 333.

the catch (which prevents the rod *R* from moving in the direction of the pull) is tripped, and the compressed spring causes the rod *R* to move instantly to the left, and this motion reverses the entire order of working, as we find now the water flowing through the water supply pipe, the valves and the "outlet *B*" into "tank 2," which is filled to the required level, and at the instant of change, we find our boiler feed-pump drawing its supply of water from "tank 1."

As the water falls in this tank, the falling float compresses the spring *S* 1 until this float descends to the fixed level, when a second catch is tripped and the first order of working is returned to.

Each time the rod *R* is moved, a counter registers one, adding it to the previous figure, and thus, by reading the counter, the number of times each of the tanks have been emptied is noted.

Each of these changes, as shown on the counter, represents

1,000 cubic feet of water, and I found this method of measuring constantly the amount of water used by the boilers much more accurate than any meter or other measuring device I could find.

The float used by me fell into a very small tank, placed at the bottom and in one corner of the large tank, this small tank being but little larger in diameter than the float itself. By this arrangement I could trip my valve by drawing out no more than an ordinary tumblerful of water. The float also rose into a similar small tank, which governed the height of the water admitted to the tank with equal accuracy. The bottoms of both of the large tanks were so arranged as to completely drain all of the water they contained into these small tanks, and this insured their being emptied entirely each time.

In calibrating these tanks and in adjusting the position of the floats so that they would change the valve after exactly 1,000 cubic feet of water passed through the valve, I used an accurate platform scale, which I wheeled directly over the tanks, and weighed out carefully the exact amount of water used between the time the water ceased to flow into the tank and when the valve changed and the pump took its supply of water from the other tank. If it was desirable to take into account the small change in the volume of the water as the temperature varied, this could easily be done, as, in the daily log that was kept, beside the number of thousand cubic feet of water used, the temperature of the entering water was noted.

This or some other accurate method of keeping a constant record of the amount of water evaporated each day by boilers is one which I believe will, before many years, be adopted by all progressive owners of large steam plants, as it will offer them a means of knowing, each day, the exact amount of power they are using, and if their coal is weighed at the same time, it will show them at once the real evaporative value of their fuel and besides afford them a means of determining for themselves the true commercial value of the coal used, which thus can be easily compared with other coals. By this means their fuel bills will be decreased, and a large amount saved every year.

This system would also aid manufacturers in making a comparison among themselves as to the relative efficiency of the different boilers they are using, and this would teach many the lesson that the cheap boiler is generally too expensive in the end to indulge in.

I found this method of measuring carefully the amount of water I evaporated a most useful check on my coal. In this plant my coal yard was located just outside of my boiler room, and all the coal was wheeled in on wheelbarrows. Each barrow was made to weigh exactly alike, and every load of coal that was run over the scale would bring the beam to a balance exactly when 300 lbs. of coal was in the wheelbarrow. The firemen soon had this matter of weighing the coal down to a fine point. They would fill their barrows to a certain point and lay the last shovelful of coal, shovel and all, on the top; when on the scale, they lifted the shovelful and scattered a few pieces of coal on the top of the load, which would balance the scale, and the rest of the coal, with the shovel, would be left outside the boiler room. Each load here was also registered on a regular counter, operated by pushing a button.

By thus obtaining a true record of the weight of coal and water used, I could soon determine the average evaporation per pound of coal, through which I was able to get at the true commercial value of my fuel.

Often a coal dealer would come in and offer me a "very fine quality of coal," and if I was in the market and was inclined to believe his statement, I would order 40 or 50 tons as a sample, and after running a few days with this coal, I could easily say whether I could afford to buy it at his price, and if so, I was able afterwards to determine accurately whether the coal he delivered was the same as the sample.

The method I have described for weighing coal is very crude compared with the one the C. W. Hunt Company, of New York, have recently put in a number of large plants. By this system the coal is first elevated higher than the tops of the boilers and is then run down into long square tubes, which are connected with a regular scale beam. A definite amount of coal is first delivered into the square tube, where it is weighed, and then, by opening a valve at the lower end, the coal runs out on the floor, near the boiler; the valve is then closed, and another charge is received in the tube and weighed, and it is held there until the valve is opened again, when the first charge of coal is exhausted.

Mr. Chas. E. Emery.—We are disappointed to find that this paper does not appear to throw any new light on this very interesting and important subject, although from the reported experience of the author, as the former engineer of a large manu-

facturing establishment, and, therefore, presumably charged with making the estimates, he should have been able to furnish the detailed information as to the first cost of steam machinery and of its erection, which it is so difficult for an engineer not actually engaged in shop practice to keep track of accurately. Instead of this, the paper starts out to make a comparison based on the cost of a pumping engine, which was necessarily very high, as the cost of the pumps was at first included; the speed was very low and the engine specially designed for the kind of work, and, although an allowance for pumps and for increased speed is made afterward, it is done by correcting the costs for the same machinery and not by stating originally the costs of machinery best adapted for furnishing power to a revolving shaft, and there are, moreover, no itemized costs, such as would naturally be expected of an engineer accustomed to make estimates. A tabular statement approximately distributing the total cost to the several items was attempted in our last paper on the subject,* and we would have been very much pleased to have had these details amended or enlarged; but this has not been done, although the author's statements follow our special treatment of the subject so closely that there is no doubt he had the paper before him.

It may be stated, also, that the cost of an indicated horse-power is not what is wanted by steam users, and that the depreciation named is too high. The lack of details makes it impossible to thoroughly check the work, but the final results indicate that some items of importance have been omitted or generalized with others, so that part of such results are too small and others too large. These various defects seriously detract from the value of the information sought to be imparted, and it is to be regretted that the author did not try to supplement information formerly published on the subject instead of attempting a generalization involving details and methods, with which he evidently is unfamiliar.

M. DeCourcy May.†—I greatly regret that I was unable to be present at the Montreal meeting of the Society, and take part in

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* (See paper on "The Cost of Steam-Power produced with Engines of Different Types under Practical Conditions, with Supplement relating to Water Power," Transactions Am. Inst. of Electrical Engineers, March, 1893. See also original paper on "The Cost of Steam-Power," Vol. XII., Trans. Am. Soc. C. E., November, 1893.)

† Author's Closure, under the Rules.

the discussion of my paper, the object of which was to lay before the Society such data as I have been able to collect concerning the costs of an indicated horse-power from actual engines running under ordinary conditions. It is gratifying to note that the two engines given by Mr. Webber agree closely with my figures, they being, as he states, good examples, and above the average. I am sorry that the paper did not interest Mr. Holloway as it should have done, but had he given more attention to the "play," I think he would have found that both the type of engines and the price of coal are given in the table printed therein. In reply to Mr. Emery I regret that he is disappointed in the lack of detail in my paper. It is satisfactory to me, however, to find that I have followed so eminent an authority's "special treatment of the subject so closely, "without having, as he supposes, seen his paper on "The Cost of Steam-Power produced with Engines of Different Types under Practical Conditions, with Supplement relating to Water Power," read before the American Institute of Electrical Engineers.

I find that the object of my paper will be attained if Mr. Emery will supplement it, and throw new light on this interesting and very important subject before the American Society of Mechanical Engineers.

DCIV.*

MECHANICAL DRAFT.

BY WILLIAM R. RONEY, BOSTON, MASS.

(Member of the Society.)

THE importance of good draft, natural or artificial, for supplying sufficient oxygen for the economical combustion of fuel, has long been recognized by intelligent engineers. The gain, both in efficiency and capacity, obtained by the rapid and energetic combustion of the various kinds of coal, and the resulting high furnace temperatures, is well established. Its importance has, however, been generally conceded only within a few years. The wonderful stimulus which the development of electrical industries has given to the building of compound engines has necessitated higher boiler pressures, and this in turn has greatly increased the use of boilers of the water-tube type. While high initial temperature in the furnace is essential to the best economy with all types of boilers, it is especially so with water-tube boilers, with their large amount of heat-absorbing surface in close contact with the products of combustion; as otherwise, the temperature of the gases will be lowered below the point of ignition and will pass to the chimney only partially consumed. To obtain this high furnace temperature requires draft sufficiently strong to deliver an abundant supply of oxygen to the furnace.

There are two well-known means used for accomplishing this result, viz.: natural draft, produced by a column of heated gases in a chimney of suitable proportions; and forced draft, obtained by mechanically creating a pressure under the grates with a blower or fan.

A third means, less widely known, is mechanical exhaust or induced draft, produced by a suction fan so arranged as to draw the waste gases from the furnace and discharge them into a short stack sufficiently high to clear the surrounding buildings.

* Presented at the Montreal meeting (June, 1894) of the American Society of Mechanical Engineers, and forming part of Volume XV. of the *Transactions*.

Although the idea is by no means a new one, yet it is only within a few years that mechanically induced draft has been much used or installed on a large scale. Previously it had been used, with a few exceptions, for the purpose of improving poor draft by helping out an insufficient or an overloaded chimney. On account of the height of the stacks being necessarily limited, artificial or forced draft has been used on ocean steamers for many years. Here the usual method of producing artificial draft is by rapid-running pressure blowers discharging under the grates, or into the fire room, which is made practically air-tight, and maintaining an air pressure in the room of 2.5 to 3 ounces. This method of producing artificial draft is at an expense for power to drive the high-speed blowers which would not be considered for a moment except on shipboard, where the limited space, as a rule, prohibits the use of large, slow-running fans, and where economy in fuel is not the most important consideration. Experiments in the use of mechanically induced draft are now being made on some of the Atlantic liners. The steamer *City of Berlin*, of the American line, was fitted up last year with large steel exhaust fans, driven by direct-connected engines, and having a capacity in cubic feet of air per minute sufficient to produce a draft equal to 3.5 inches of water. Connections were made to the smoke flues, and the gases drawn from the furnaces through a by-pass and discharged into the steel smoke stacks. The results obtained are not yet made public, as they are still being tested experimentally.

The largest and most successful applications of mechanically induced draft have been made in connection with feed-water heaters designed to utilize the waste heat of the flue gases, and known as fuel economizers. This form of feed-water heaters has been manufactured in England for over fifty years, and in this country for three or four years. They have, however, been imported for many years, as their value as a fuel-saving device is well established. Their successful operation is however so dependent upon good draft, that no well-informed engineer would think of installing an economizer without making provision for much better draft than the boilers would require without it.

On account of the reducing effect on the draft caused by lowering the temperature of the gases and retarding their flow by the mechanical interference of the pipes, it cannot be considered good engineering to attach an economizer to a chimney less than

200 feet in height. In fact, the best working economizers in connection with chimneys are those where the chimney is considerably over 200 feet high.

The objections to be urged against high chimneys, as compared with mechanical exhaust draft when used with economizers, are: First, excessive cost, both on account of the height required and on account of foundations, which must of necessity be very substantial, and which may involve expensive piling and filling. Second, the space required for foundations, which may be very valuable, especially in large cities, or may be required for other purposes and which can with difficulty be spared. A chimney 250 feet high will require foundations not less than 30 feet square, and in some cases much more. Third, a certain minimum temperature of flue gases is required to produce an effective draft and to operate the boilers economically, and this fact limits the amount of economizer heating surface which can be used, and consequently the fuel saving obtained by use of the economizer. The same fact operates unfavorably at small capacities, which are often unavoidable, when the chimney must be built large enough for future increase of the boiler plant. Fourth, a chimney once built limits the maximum capacity of the boiler plant, and also is liable to be affected by atmospheric changes which may seriously impair its efficiency.

These objections to the tall chimneys which are so essential to the use of economizers, do not hold with mechanical draft.

The first cost of a properly designed mechanical draft plant, exclusive of economizer, is very much less than that of a suitable chimney of equal capacity, often averaging 50 to 75 per cent. less, according to the size of chimney and character of foundations required. The fans and short stack require very little foundations, even less than that of an ordinary boiler setting. The space usually required for extensive chimney foundations can be utilized for economizers, and by elevating the economizers and fans upon beams and columns the space underneath them can be used for pumps, condensers, etc., as indicated on Fig. 334, showing "cross section of a typical boiler-house." The space thus saved is often of great value, especially where land is expensive.

Natural draft requires that the gases in the chimney be above a certain minimum temperature in order to secure a proper supply of oxygen in the furnace and a good combustion of the

fuel, whereas with mechanical exhaust draft the amount of draft obtainable is entirely independent of the temperature of the flue gases, and when used in combination with a properly proportioned economizer, it is possible to lower their temperature to a point where the draft of even a very tall chimney would be practically destroyed.

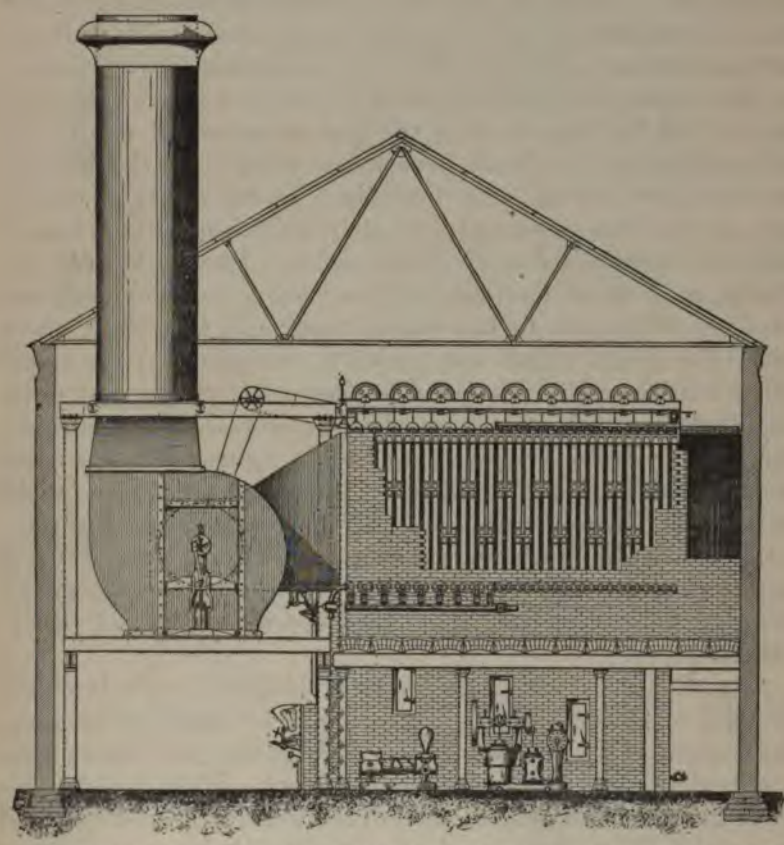


FIG. 334.—Cross section of a typical boiler-house, showing boilers, stokers economizer, mechanical draft, feed pump, and condenser.

Mechanical draft possesses great advantages over natural draft in its flexibility and adaptability to both large and small capacities, and in its ability to meet sudden and excessive demands for steam, either by an extra turn of the throttle valve, or by an automatic regulator controlling the steam supply to the fan engine according to the boiler pressure. It is unaffected by

atmospheric changes, furnishing the desired amount of draft irrespective of conditions of wind or weather. Operating independently of the amount of heat in the stack, it is possible to obtain a higher temperature of feed-water in the economizer, and a lower temperature of escaping gases, than could possibly be obtained with a chimney, and at the same time provide sufficient draft to maintain rapid and economical combustion of the fuel. There are undoubtedly many boiler plants equipped with economizers and chimneys, where the draft is so greatly reduced by the economizer, that it is an open question whether the saving in fuel by thus heating the feed-water is not more than balanced by the loss due to imperfect combustion in the furnace; and whether it would not result in a greater saving in coal to cut out the economizer and get better combustion, and a higher initial temperature due to better draft. Unquestionably the "black eye," which fuel economizers have sometimes received, has been often due to bad engineering, and to placing them where the chimney draft was none too good already; the result being, that they not only failed to show the economy which the purchaser expected, but so impeded the draft that the efficiency and capacity of the boilers were greatly impaired. It was quite natural, under such circumstances, that the economizer should be neglected and allowed to foul up by the accumulation of sediment within the pipes, and of soot without, until it became a source of loss instead of economy. A chapter on the use and abuse of fuel economizers could easily be written, but—"that is another story."

A mechanical draft plant properly designed, with duplicate fans and engines of suitable construction, so arranged that one is always in relay, can be made so reliable that the boilers cannot be shut down by any ordinary accident. With the fans properly designed and proportioned to the work, the power required to operate them is so small as to practically have no effect on the economy obtained. The mistaken idea that prevails somewhat, even among intelligent engineers, regarding the amount of power required for mechanical exhaust draft, is probably caused by the well-known large amount of power required to drive the high-speed pressure blowers and fans used for forced draft. Mechanical draft handles a large amount of heated gases with slow-speed exhaust fans at a low pressure, and with a small expenditure of power. To illustrate:—the writer re-

cently designed a mechanical draft and economizer plant for 6,000 H.P. of water tube boilers, providing duplicate large slow-running fans of special design, each driven by an independent engine, and each having a capacity, estimated in pounds of coal burned per hour, sufficient to develop 25 per cent. in excess of rating, or 7,500 H.P. The power required to drive one fan to do this work was *six-tenths of one per cent* of the boiler horse-power developed. Or, estimated in coal per horse-power per hour, and at $3.00 per ton, the fuel cost of operating the plant one year was *two per cent.* of the estimated cost of the chimney originally planned for the plant. In other words, *it would not pay to build the chimney so long as money was worth more than two per cent. per annum.*

An illustration of a typical boiler house, showing boilers, stokers, economizer, mechanical draft, feed pumps, and condenser, as shown in Fig. 334, will perhaps be of interest. In this illustration the economizer is elevated upon columns and beams to provide for utilizing the space under the economizer for feed pumps, condenser, etc. The exhaust fans, of which there are two placed side by side, are equipped with direct-connected engines, only one engine showing in the illustration, the other being on the farther side. These fans and engines are of special design, with protected bearings, self-oiling and water-jacketed, to withstand the heat when the economizer is cut out for cleaning or for repairs, and the hot gases pass directly to the fans. They are so proportioned to their work as to handle a maximum amount of gases with a minimum expenditure of power.

The arrangement of the economizer pipes and blow-off connections is worth noticing, in that they provide a means of blowing out the sediment which may accumulate in the pipes, and at the same time a complete circulation is maintained in the economizer. Many extensive plants are now in operation, or in process of construction in various parts of the country, equipped with economizers and mechanical draft similarly arranged.

Figs. 335, 336* and 337, show longitudinal section, cross-section and plan of probably the largest plant of the kind yet built; viz., that of the Philadelphia Traction Co., Philadelphia, Pa. Two large power-houses are in process of erection and partly in operation for this company; one of 7,500 H. P. and one of 6,000 H. P. The illustrations show the arrangement in the Thirteenth

*From drawings kindly loaned by *Power*, New York.

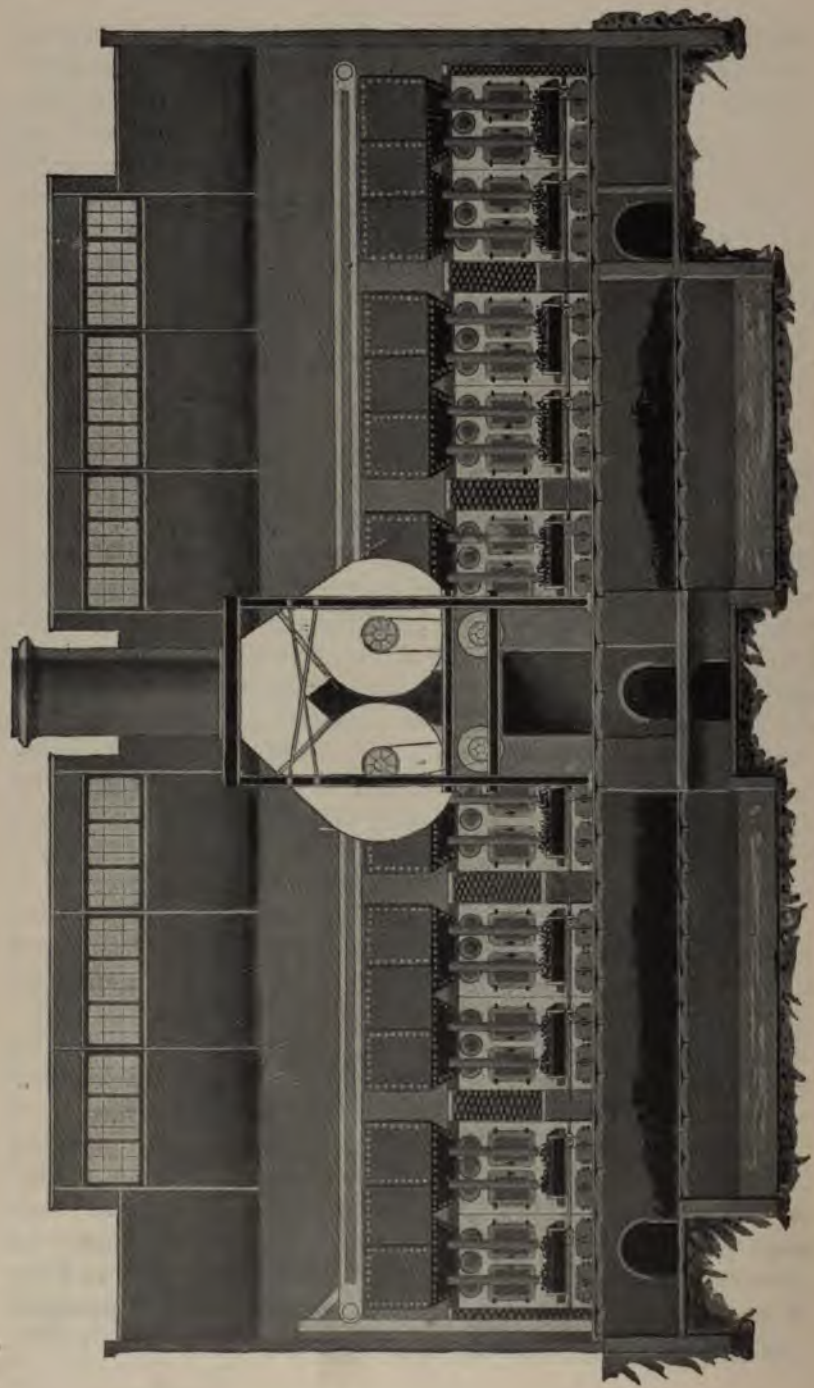


FIG. 335.—Longitudinal section of boiler-room, Philadelphia Traction Co.

and Mt. Vernon Street Power House, where there are twenty Babcock and Wilcox boilers of 375 H. P. each, arranged in two parallel rows of ten boilers on opposite sides of the boiler-room. The other station of 6,000 H. P. is arranged similarly, except as to the number of boilers. The gases from each row of boilers are conveyed by flues at the back to the centre of the

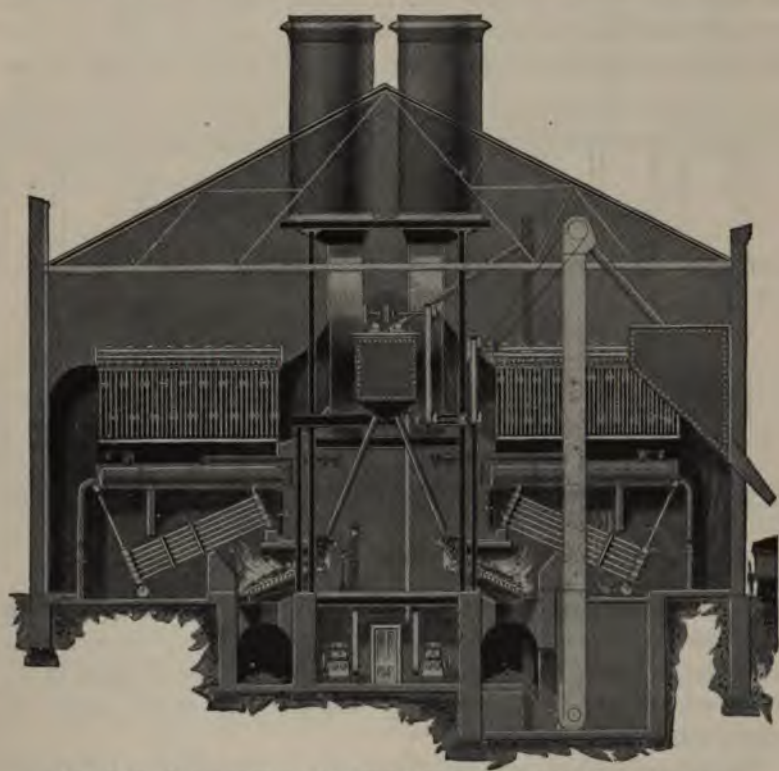


FIG. 336.—Cross-section of Philadelphia Traction Co.'s boiler-room.

boiler-house, as shown on plan view, Fig. 337, drawn through the economizers and discharged into stacks by four large slow-running exhaust fans of special design. These fans are arranged in pairs, and are of such a capacity that two of them will handle the gases for the entire plant, thus leaving two in reserve. The fans are driven by duplicate engines and counter-shafts, so arranged that either engine will drive any one or more fans, as desired. The stacks extend but a few feet above the roof, and are lined with brick to preserve them from the corroding action

of the gases. The feed-water is pumped through exhaust-steam heaters to the economizers, and thence to the boilers. At the Thirty-third and Market Street station of the same company, the fans are driven by engines belted directly, one small engine to each fan. The economizers as shown on plan view, Fig. 337, occupy the space originally planned for two tall chimneys, one for each side of the boiler-house. These chimneys were to have been 200 feet high with a flue eleven feet in diameter. Two of the fans are capable of producing a draft equal to that of the

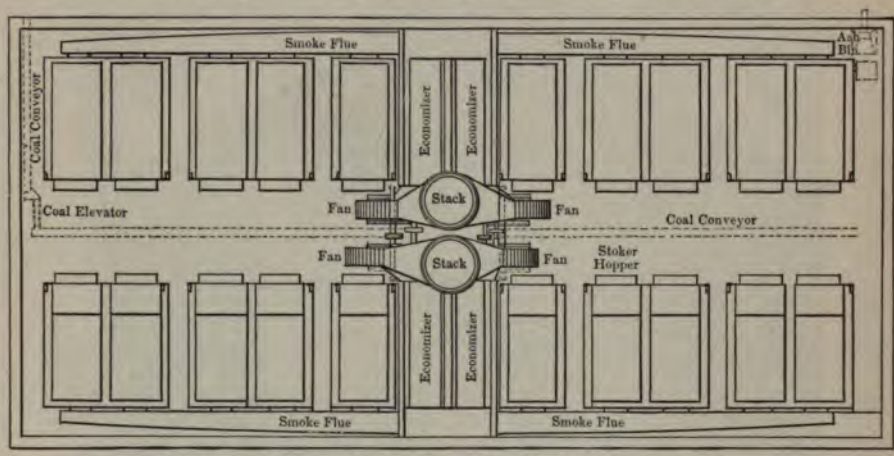


FIG. 337.—Plan of boiler-room, Philadelphia Traction Co.

two brick chimneys originally planned. The power required to drive one of these fans, as the plant is now being operated, is exceedingly small, being less than ten H. P. for each 2,000 H. P. produced, or *less than one-half of one per cent of the power developed by the boilers.*

The following data will be of interest, as showing in tabulated form the results obtained by economizers and mechanical draft in a number of plants in regular service. In every case the feed-water was partially heated by exhaust-steam heaters, or in hot wells by condensed steam from various sources.

TESTS OF ECONOMIZER AND MECHANICAL DRAFT PLANTS, SHOWING INITIAL AND FINAL TEMPERATURES OF FLUE GASES AND FEED-WATER IN DEGREES FAHRENHEIT.

Plants tested.	Gases entering economizer.	Gases leaving economizer.	Water entering econom.	Water leaving economizer.	Gain in temp. of water.	Fuel saving, per cent.
1	610	340	110	287	177	16.7
2	505	212	84	276	192	17.1
3	550	205	185	305	120	11.7
4	522	320	155	300	145	13.8
5	505	320	190	300	110	10.7
6	465	250	180	295	115	11.2
7	490	290	165	280	115	11.0
8	495	190	155	320	165	15.5
9	595	299	130	311	181	16.8

DISCUSSION.

Mr. Albert A. Cary.—I believe that I have already spoken, before this Society, of an apparatus somewhat similar to the one just described by Mr. Roney, which I designed and used in 1887; and this device was certainly an original conception, and, as far as I can learn, it was the first one of the kind ever constructed and operated.

I remember going to the Sturtevant Fan Company, when working on this mechanical draft, for a fan. They refused to build it for me, as they said it was impracticable, and that a fan could not be put to this use, handling hot gases, successfully; but now I believe that they are building many fans for this purpose; and the great value of an induced draft accomplished by this means is beginning to be quite widely appreciated, thanks to the careful experimenting and able handling of this device by Mr. Roney and his associates, Messrs. Westinghouse, Church, Kerr & Co.

In the steam plant where I introduced this device, in the early part of 1887, I found that the demand for steam at times was greater than the supply, notwithstanding the economy that was practised in its use. The boiler-room had but one vacant space, and this was comparatively small, where but one boiler could be placed, which alone would not give sufficient steam. I found that, when the factory was running regularly, the escaping gases from the boilers ran from 650° to 700°, so, seeing a chance for

saving here, I decided that I would put in a Green Economizer in connection with a new boiler. Now another difficulty confronted me, and that was, with a chimney already too small without the extra boiler, how could I get along, especially after using an economizer and lowering the temperature of my gases some three or four hundred degrees, thereby reducing the efficiency of the chimney still further?

To build a new chimney, in addition to purchasing a new boiler and an economizer, was altogether too expensive to consider, to say nothing of the inconvenience of a "shut down" while the old chimney was torn down and the new one erected.

I finally made up my mind that a forced draft would have to be used, whereupon I visited many plants using forced draft, and examined the devices they employed.

All were forcing air under their grates, employing no small amount of power to drive their fans, and burning their grates and furnace linings out in very short periods of time.

The best arrangement of this kind I found was in ocean steamship practice, where they made their stoke-holes air-tight, and kept the air in them under pressure; but this system could not be used in my case, so I finally came to the conclusion that I was looking in the wrong direction. I had used a steam jet in the chimney, but experience proved to me that it was more a device to "rob Peter to pay Paul" than it was the one to solve the problem I had under consideration, as it required a great deal of steam, while I had no steam to spare; but, nevertheless, it increased my chimney draft, and that was what I wanted. This led me to think of putting a fan between my boilers and stack, arranged so as to exhaust the air from the furnaces and force it up the chimney; as I knew from previous experience with fans that the steam required, when the power was taken from the large Corliss engine (driving one of our mills), would be but a small fraction of that required by the steam jet in the chimney.

Then, by further reasoning, I thought that it would be a matter of economy to increase the size of my economizer so as to take care of the higher temperature of flue gases brought about by the increased combustion per square foot of grate surface, through the induced draft, and lower the temperature of these gases as much as possible; as my chimney draft would, with this device, be entirely independent of the temperature at its base;

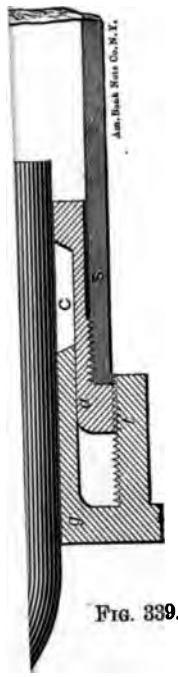
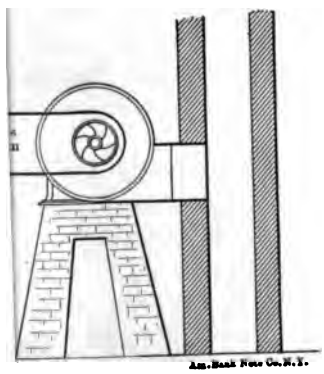


FIG. 339.



and I thereupon made designs for this plant, based upon these theories.

In my first experiments with fans I used those of the propeller type, such as are used for ventilating; but as the area of my chimney was too small for the service, I found that I could not get pressure enough with them to expel the large quantity of gases they had to handle; and in order to obtain this pressure I was finally obliged to resort to a pressure blower.

As my arrangement differed in many respects from the plant described by Mr. Roney, I will make a sketch of it (see Fig. 338), from which I can describe it plainly.

The arrangement of the five horizontal return tubular boilers and the one vertical boiler is too plainly shown in the plan and in the front and side elevations to need description. The Green Economizer is also clearly shown, placed directly over the rear of the horizontal boilers. Directly under the floor of the economizer is placed the main flue, which collects the discharged hot gases from all of the boilers.

At *D* and *D'* doors or dampers are shown, which are hinged at *h* and *h'* respectively. By arranging these in the positions shown, the hot gases must pass along the main flue, from right to left, and then upward into the economizer, where they pass among the many vertical tubes, giving up their heat to the feed-water on their way; and on arriving at the extreme right end they plunge downward and turn through the wrought-iron flue towards the chimney.

By moving the two dampers through an angle of about 90° (as indicated by the dotted lines), you will see that the economizer will be cut out, and the hot gases will pass directly (from left to right) into the wrought-iron flue connecting the main flue to the fan and chimney.

The position of the fan is too plainly indicated to need description, and as it revolves it draws the hot gases in towards its centre, on each side, and discharges them from its outlet into the chimney. It is needless to say that the faster the fan revolves the greater the "suction" or draft.

Thus far but very little difference will be seen between this device and the one described by Mr. Roney, but I will now proceed to show wherein it lies.

In the fans used by Mr. Roney, I believe that the bearings are kept cool by a circulation of water around the journals.

In my device I used, instead, a large solid-drawn steel tubing about six inches in diameter, which I obtained from Messrs. Leng & Co., of New York city.

A rapid stream of cold water passed constantly through this hollow shaft, keeping the bearings perfectly cool, and as this water was afterwards used in the boilers, the little heat absorbed was thus not wasted.

The method I used in connecting my delivery and discharge water-pipes to this shaft is shown in section in Fig. 339.

A brass stuffing-box (*a*) is first screwed into the end of the shaft (*s*). After the packing is placed in the chamber (*c*) around the pipe (*p*), the gland (*g*) is screwed into place, as shown, and made tighter by screwing it down on the screw (*t*).

This I found a very effective and tight joint, which allowed the pipe (*p*) to remain stationary while the shaft, with its end fittings, revolved.

One of the points of interest in this device was the method I used to vary the speed of the fan according to the demand made upon the steam.

By examining the plan view of this plant you will see that the fan received its motion through a pair of speed cones, and the speed of the fan, of course, varied according to the position of the belt connecting these two cones.

Fig. 340, an enlarged detailed view, will show the method used to accomplish this change of speed in the delivering cone.

A screw was placed above these cones, as shown, which carries a nut, which is prevented from revolving with the screw by contact with a parallel rod. This nut carries a "belt shipper," which consists of two rods dropping down and inclosing the belt (which connects the cones) between them, thus: when the screw is revolved in one direction the nut moves towards the right, carrying the belt with it, and thereby increasing the speed of the delivering cone. Necessarily, when the screw is revolved in the opposite direction, the belt moves towards the left, and the speed of the delivering cone decreases.

To accomplish this end, a rectangular framework, such as is shown in Fig. 340, was constructed, and supported on the left-hand end of the long screw.

This support formed an axis on which this framework vibrated, or performed a fractional part of a revolution.

At the left end of each of the two cones, friction disks were

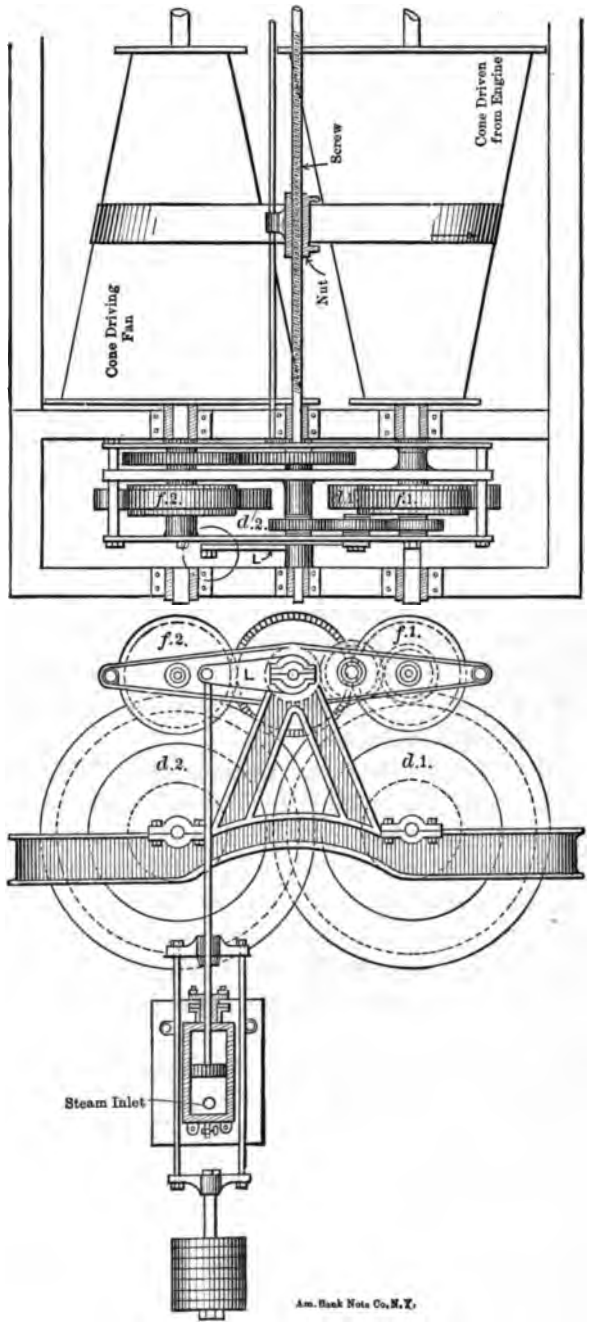

Cone Driven from Engine
Screw
Nut
Cone Driving Fan
f.2.
f.1.
d.2.
d.1.
L
Steam Inlet
Am. Bank Note Co. N.Y.

FIG. 340.

placed, so as to be directly under two friction wheels (f 1 and f 2) in the rectangular frame.

When the frame was tipped so that the friction wheel f2 was brought into contact with the friction disk d 2 (supposing the belt to be running in the direction indicated by the arrow), it would communicate its motion through the gearing to the screw, so that the top of the screw would move away from the cone in contact. Of course, when the other friction wheel (f1) was brought into contact with the other friction disk (d 1), the direction of revolution in the screw was reversed.

The oscillation of this framework carrying the friction wheels was accomplished by means of a Hallock damper-regulator, as shown clearly in the side elevation, Fig. 340.

This regulator consists of a cylinder, having a piston which moves up and down. The pressure of the steam is exerted against the under side of the piston, which raises it and its piston-rod, and this is counterbalanced by weights, which are suspended beneath, as shown.

When the pressure of the steam exceeded, say, ninety pounds, the piston was raised, pushing the piston-rod and lever up, through a limited distance. This, of course, brought the friction wheel f1, and the friction disk d 1, into contact, and the speed of the fan was reduced.

On the other hand, when the pressure was less than 90 lbs., the weights caused the piston to descend, and the other friction wheel and disk were brought together, which quickened the speed of the fan, and by this means the greatest uniformity of steam-pressure was obtained; and as the draft was limited by the demand for steam, a considerable economy resulted in the consumption of the fuel, as the combustion of coal, of course, is limited by the quantity of air passing through the grate bars.

While the factory was running regularly the power to drive the cones was taken from the large engine, but, when this was not running, a small auxiliary engine was thrown into service by means of friction pulleys.

One other somewhat novel idea was employed by me in this plant.

It consisted of a device for closing the damper in the individual flue of each boiler, when the fire door was opened for stoking or cleaning.

It was invented, I believe, by Mr. John Ashcroft of New York city, and is illustrated in Fig. 341.

The door on the right, of the pair shown, lapped over the one on the left, so that it was necessary to open it first. A piece of gas-pipe passed through the hinges, and turned with the door. At the upper end of this piece of pipe a lever was fastened, and a connection ran from this lever back to the damper, and thus the door, gas-pipe, and lever, all moving around together, caused the damper to close, and the great influx of cold air was thus avoided, with all its attending evils.

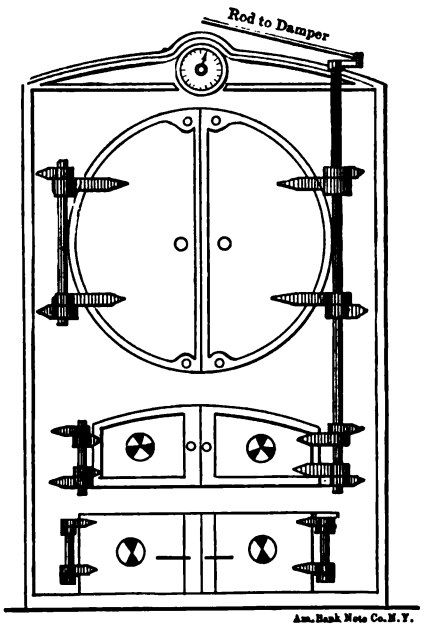


FIG. 341.

As the damper device was carefully balanced by weights, the opening of the door was a very easy operation.

The President.—I do not think there is anything before the profession more important than this very matter. You can hardly make people believe the immense amount of money wasted in chimneys. The question of the cost of running a chimney was discussed at great length and with great ability by Prof. Benjamin W. Frazer of Lehigh University, in a paper in the Transactions of the American Institute of Mining Engineers some ten or fifteen years ago, and there he says—not only says but demonstrates, after a very careful analysis of the question—that the most expensive way of getting a draft is a chimney, even if somebody gives you the chimney; and in our own works we only have about 12 feet of the chimney above where the feed-water heater which is in the stack ends; and in certain of our works we have what I think in one of my papers I said that one dude called another—a combination of a "blower" and a "sucker;" that is to say, we force the air hard enough just to make it get through the fuel, so that above it we have practi-

cally neither a plenum nor a vacuum, and then we suck it through the furnace. We find there is a certain velocity which we want, and if we go above it we get carbonic oxide in our stack and have the stack too hot. If we go below it, we get inefficient combustion; and my belief is, just as Mr. Roney says, that you want to control absolutely, and perhaps automatically, the flow of your waste gases through the furnace.

Mr. Geo. I. Rockwood.—There are two purposes for which a chminey is used in stationary plants; one is to create a sufficient draft, and the other is to get rid of obnoxious gases. This last object is attained only by giving the chimney sufficient height above the surrounding buildings, as no amount of air pressure will hurl the products of combustion vertically against a slight wind pressure across the top of the chimney. Often, in order to properly rid the surrounding buildings of soot, etc., the formation of which the best automatic stokers do not entirely prevent, it is necessary to carry up the top of the chimney to a considerable height, as, say, from 100 to 150 feet in manufacturing districts, and very much higher in thickly settled city districts. I refer to this because Mr. Roney advocates (see page 1164 and illustrations) the use of a short steel stack, only high enough to clear the roof of the boiler-house, and because there are several large boiler plants with induced draft and excessively undersized chimneys in operation about the country, which, in this respect, stand for anything but arguments in favor of the substitution of artificial draft for natural draft.

The other purpose of the chimney—the creating of a sufficient draft—is as often fulfilled as a result of carrying up the chimney to a proper height to get rid of the gases as not. The cases where a chimney would naturally be built lower than a height great enough to afford proper draft are, I believe, rather infrequent, and are confined to very large plants, such, for example, as the plant described in the paper; but even in this case I confess it seems to me injudicious to curtail the stack quite so much as shown in the illustration. Where, however, it is possible to use the short stub of a chimney and the suction fan connected in the manner shown, especially where the plant is a large one, requiring otherwise an enormous brick stack, the illustrated design is undeniably a very clever and excellent substitute, even in the matter of cost, while there is the great advantage of a draft controllable at will.

The question of the wisdom of employing mechanical draft is not, however, to be confounded with that of the wisdom of introducing feed-water flue-heaters into the path of the waste gases, although the two matters are closely related in the paper. It is not at all clear in my own mind that the flue-heater is necessarily, in new plants, a wise investment, though I am open to conviction on this point. I would feel obliged to Mr. Roney if he would add to his interesting paper tables showing the ultimate saving in money due to the addition to the plant of the heater, after paying the two items of interest on cost of heater, setting, and ground occupied; of outlay for repairs and extra attendance; reckoning coal at $4 per ton. I will calculate the saving which it appears to me would be realized in the case of an ideal plant having compound engines of say 1,000 H.P., and working under the best conditions. Suppose steam to be made at a pressure of 130 lbs. in the boilers. It would be the result of poor design and upkeep if the temperature of the flue-gases were as high as 610°, the highest figure given by Mr. Roney in the table. Under the best conditions that temperature should be not over 430°. Now the fuel saving due to the flue-heater will depend upon the increase in temperature which it gives to the feed before the water passes into the boilers. If there are any non-condensing engines or pumps at hand, their exhaust may be utilized in an ordinary pressure heater to raise the temperature of the feed from, say, 90° to 212°. If no exhaust steam can be thus utilized, then steam may be taken from the receiver of the compound engine for this purpose. (By the way, it will not do to use a pressure heater in connection with the receiver, as it is impossible thus to save the latent heat of the portion of steam passed through the heater. There is a better type of heater for this purpose, known at sea as Weir's Feed-water Heater and on land as Webster's Vacuum Economizer. The principle of this heater is that of the jet condenser, and the condensing effect of the feed-water is that which causes the flow of steam into the heater without need of an opening through it to the atmosphere. The intimate and direct mingling of the steam with the water causes the complete absorption by the latter of the sensible and latent heat of whatever steam flows to the heater. Such steam, therefore, works through a perfect cycle, there being no expenditure of heat for any other purpose than that of making power in the high-pressure cylinder. If it were

passed along from the receiver into the low-pressure cylinder instead of into the heater, it would finally exhaust as steam into the condenser, and the principal part of its heat—that latent—would go off to the overflow. Looking at the matter in a different way, suppose one-half of the total work to be done in each cylinder of the compound engine; then the quantity of steam abstracted from the receiver for use in the heater would be saved, but one-half of its total capacity for doing work would be sacrificed. Consequently but one-half of the apparent saving due to heating the feed-water from 90°—temperature of hot well—to, say, 220°, the temperature of the steam in the receiver, or through 130°, would be really lost by reason of the sacrifice of work done in second cylinder. If 12° rise in temperature corresponds to a saving of 1%, then the Webster Economizer will save 5½% of the total coal.)

As we have assumed that the boilers operate compound engines, the economizer can get water at 220° and heat it to, say, 260°, as we have assumed 430° as the temperature of the waste gases. The difference is 40°, which corresponds to a saving of something more than 3%. This, then, unless I have assumed conditions impossible of fulfillment, is all that can be relied on as the saving due to adding the economizer to the equipment of the plant, if the plant is worked to the best advantage. The question I ask is: what proportion does the cost of the 3% of coal saved bear to the interest on the first cost of the economizer, plus the cost of keeping it clear of soot, plus the cost of increasing the draft to the extent which it is reduced by the action of the economizer, plus the interest on the ground occupied by it?

Of course, I am aware that the table of savings given by Mr. Roney is correct, but does it represent conditions sometimes to be met with in existing plants of overworked boilers, foul with soot, or is it really representative of the best possible practice?

Mr. Roney.—I think that Mr. Rockwood misunderstood entirely the purpose of my paper. It is to show the advantages of mechanical draft as compared with chimney draft, both in first cost and in efficiency. This is especially evident when economizers are used; and the question he raises, whether it will pay to put in economizers or not, depends upon what you pay for your coal, and how much you burn per annum. A

boiler plant can be so small, or the price of coal so low, that the saving by the economizer will not pay interest on its cost.

The President.—Another very important point is whether you have to pay a high price for stoking. The amount of that is a question to which I do not think sufficient attention is often paid. That is, in large cities where there are large unions, the firing is expensive in proportion to the amount of coal you put in. But where you can use automatic stoking, it probably is entirely different, because there, roughly, the cost of stoking is not so much a function of the cost of labor.

*Mr. Roney.**—The advantages of automatically controlling the draft are well shown in a mechanical draft plant which I designed for use in a distillery in Peoria, Ill. The fan was driven by a direct-connected engine, and the steam supply to the engine controlled by a lever valve connected to an automatic pressure regulator. The effect of automatically controlling the speed of the fan, and thereby the amount of draft, was at once seen in the uniformity of the steam pressure. Before the mechanical draft was installed, the recording steam-gauge in the still-house showed a variation in pressure often as great as forty pounds, due to sudden and excessive drafts of steam for the cookers. After the fan was put in operation the greatest variation in pressure for a period of thirty days was five pounds. The benefit of this uniform steam pressure was apparent in increased product and greater ease in working. The effect was also so noticeable in the fire-room that the proprietors thought the firemen were having too easy a time. They were found sitting around, and not watching the steam-gauge as closely as they were obliged to before the mechanical was put in.

Mr. Cary has given a very interesting account of an ingenious combination of a number of well-known elements necessary to an economical plant. He is in error, however, in thinking that the combination he used in 1887 "was the first one of the kind ever constructed and operated." Over twenty years ago Messrs. J. Edwards & Son, Halifax, Yorkshire, England, installed a Green Economizer in connection with eight large boilers, the draft being induced by an exhaust fan, which discharged the waste gases into an iron stack twenty feet high above top of fan. This bit of mechanical

* Author's Closure, under the Rules.

draft history, which I came across when preparing my paper, is one more illustration of how easy it is for engineers thousands of miles apart, when seeking to obtain a certain result, to adopt substantially the same means for its accomplishment.

In conclusion, I venture the prophecy that the time is not far distant when tall chimney building will be one of the lost arts, and the progressive engineer will point, as his monument, not to the tall and expensive chimney of the past, but to the comparatively inexpensive and economical mechanical draft of the future.

DCV.

MEMORIAL NOTICES OF MEMBERS DECEASED DURING THE YEAR.

GEORGE H. BABCOCK.

[NOTE.—A memorial monograph of Mr. Babcock, a former president of the Society, with portrait, will be found at page 636 of this volume.]

FRANZ GRASHOF.

(Honorary Member.)

WE have the sad duty of chronicling the death, on October 26, 1893, of a distinguished colleague, an honorary member, Franz Grashof, than whom none is more distinguished or beloved on either side the Atlantic. His work has made him immortal; his character has given him many friends, in whose memories he will always be preserved. He was born at Düsseldorf, July 11, 1826, the son of Prof. Karl Grashof. He was educated in the Gymnasium and Realschule of his native city, in the Gewerbeschule of Hagen, and, from 1844, in the Gewerbeinstitut of Berlin, in which latter college he studied mathematics, physics, and machine construction as specialties. He served in army and navy from 1847 to 1851, obtaining in the three years of included sea-service a valuable practical knowledge of the working of the marine engine, and of the operation of steam machinery at sea. He continued his studies at Berlin between 1851 and 1854, when he commenced teaching in the preparatory schools, conducting work in mathematics, mechanics, and machine construction. On the first of October, 1851, he became an instructor in the Gewerbeinstitut of Berlin, and thenceforward rapidly gained standing and prestige in technical work. He went to Karlsruhe in 1863, succeeding Redtenbacher, and gave his time to instruction in the school of machine construction. He held the directorship of the Technical High School, Karlsruhe, from 1867-8 to 1885-6, inclusive. In 1868 he received a call to Aachen; he was called twice to München, and received many honorable titles and invitations to high positions in his chosen field of work

From 1882 Grashof was invalided and his work seriously interrupted. He finally died, October 26, 1893, leaving a widow—*née* Henriette Nottebohm—and two children.

Grashof, during his years of vigor and health, was a voluminous writer and author. His works were distinguished by their learning and exactness, and their firm foundation upon the principles of pure mathematics and mechanics. His largest works were his *Theoretische Maschinenlehre*, his *Theorie der Elastizitat und Festigheit*, and his *Resultate der mechanischen Wärmetheorie*. He was one of the founders of the Verein Deutscher Ingenieure (1856), and a member of a number of other learned societies, including the American Society of Mechanical Engineers, of which he was made honorary member, Nov. 6, 1884.

Dr. Grashof was a distinguished man of science, a learned member of the profession of engineering, a famous investigator, a great author, and a most successful teacher; but his personal character, his noble aspirations, his kindly disposition, his loyalty to his friends, his fair and kindly criticisms where criticism became an unavoidable duty, his loving and lovable nature—these were, after all, what gave him that affection and respect, on the part of pupils and friends, which were his most comforting experiences during life, and which will long remain in the hearts and memories of all who knew him. The profession of engineering also has lost in Grashof one of its greatest exemplars.

JOHANN BAUSCHINGER.

(Honorary Member.)

Prof. Johann Bauschinger was born June 11, 1834, at Nuremburg, Germany, of humble parentage. At an early age he gave evidence of aptitude for mathematics and the natural sciences. He graduated from the Polytechnic School of his native place with honors, and thence went to attend the University of Munich, where he devoted his attention to mathematics, physics, and astronomy. In 1856 he was called to teach in the Polytechnic School at Augsburg, but the following year was appointed instructor of mathematics and physics in the Royal Trades School at Fürth. Since 1868 he has been a member of the faculty of the Technical High School at Munich, where he occupied the chair of mechanics and graphical statics, and was

charged with the erection and direction of the physical testing laboratory. While at Fürth he developed literary activity, especially contributing papers to the technical press on "practical mechanics" and the "mechanical theory of heat," and also a work, published independently, on "popular mechanics." In 1871 he published a work on "graphical statics" which has had several editions and has been translated into Italian and Russian. It is noted for its lucid and intelligent expositions. However, his most important literary works are undoubtedly his technical investigations, and his very first work in this field characterizes him as an exceedingly circumspect, skilful, and painfully conscientious observer, who reached his aims with unusual patience and pertinacity.

The investigations described in this first work, entitled *Indicator Tests of Locomotives*, were commenced in 1865, and he completed them under great difficulties while at the same time attending to his classes. This work, which appeared in vols. xiii. and xiv. of *Der Civilingenieur*, has been published separately and is a classic on the subject at the present day, and attracted the general attention of the technical classes to the author. But Bauschinger's main field of investigations was "testing of materials," and in connection therewith his name will be mentioned for years to come. The results of these researches were at first published by the Bavarian Association of Architects and Engineers, but later in the *Mittheilungen aus dem mechanisch-technischen Laboratorium der Hochschule Muenchen* (Communications of the Mechanico-technical Laboratory of the Munich High School).

To Bauschinger belongs the credit of having called into existence the first modern public testing laboratory in Germany, which has since served largely as a model for most of the subsequent establishments of the kind.

He has developed our methods of testing in a material manner, by devising delicate auxiliary measuring apparatus. For example, his application of the Gauss method of mirror measurements has enabled us to observe behavior of materials in an absolutely accurate and infinitely delicate manner. By his mirror-apparatus, elastic deformations of the heaviest parts of machinery or bridges can be most readily observed, and it is due to him that the work of German investigators has become the most reliable and accurate of any.

His investigations of the properties of cements, mortars, artificial and natural building stones, are given in Parts 1, 7, 8; 4, 5, 10, 11, 18, and 19 of the *Mittheilungen*. In the former he treats of cements and cement and lime mortars, and in the latter of artificial and natural building stones; he describes various methods of investigations, and compares them in a most admirable manner. His especially numerous tests of elasticity and resistance of building stones must be described as "classical." The use of mirror-apparatus, of spherical supports, the elucidation of effect of friction of supports, the necessity of greater distance between end bearing surfaces from gauge marks, the effect of soft bedding strips (lead, rubber, and paper) for these surfaces in compression tests, have been presented in such a masterly manner that it is a pleasure to follow his demonstrations.

In Part 6 of "*Mittheilungen*," Bauschinger treats of the laws of resistance to compression by first reviewing the earlier work of English and French investigators, and then adding his own tests and experiments. He at first confirms the law of similarity, in accordance with which similar bodies of the same material suffer similar deformation under equal loads, brittle bodies showing equal resistance to crushing, and then he determines the laws which govern relation between the ultimate resistance and length and shape of cross section of the compressed material.

The effect of decreased bearing surfaces or eccentric loading is studied, as well as the appearance of fractures, in which particular reference is had to the pyramidal fractures in compression tests.

Parallel bending, tension, compression, and shearing tests of each material have been studied most exhaustively. For determining wear or abrasion of stone a special method is devised; and the methods of determining permanency of stone under effect of frost and weather are compared and very much simplified.

In connection herewith his investigations of strength of iron and stone columns and their resistance under the action of fire, as reported in Parts 12 and 15 of the *Mittheilungen*, must be mentioned.

Just as these investigations have given a remarkable impetus to similar work by others, his circumspect and careful tests of

timber described in Parts 9 and 16 have been followed by later investigators very closely.

Parts 2, 3, 13, 20, and 21 treat of the properties of metals, as does also a paper on "The Properties of Iron and Steel," containing the numerous investigations carried out by order of the Association of German Railroad Managers; and his report on "Tests of Resistance of Cast-iron, Iron, and Steel of the Reschitza Works," prepared for the Paris Exposition of 1878 by request of Austrian state railways.

We may here mention his reports in Part 13, on "Changes of Elastic Limit and Resistance of Iron and Steel due to stretching, crushing, to warming and cooling, and to oft-repeated loading," as classics on the subjects treated. By these we are enabled to trace the connection between the usual tests of resistance of structural materials and Woehler's tests of repeated loading, as Bauschinger's laboratory contained a full set of the necessary machines for such investigation, and he devoted many years to the study of their results. At the time of his death he was engaged in further study and investigation of the effect of continued use or loading, and in future very much greater attention will be devoted to this subject, especially should his successor carry out the work on Bauschinger's plan and methods.

And last, but not least, must be mentioned the fact that it was mainly due to Bauschinger's energy, tact, and indefatigable labors and uniform amiability, that the now famous "Conferences on Unification of Tests and Methods of Testing Structural Materials" have been so eminently successful. Had it not been for his guiding hand, constant vigilance to avoid dangerous discussions and to straighten out tangled arguments, it is doubtful whether so much could ever have been achieved, with a voluntary assemblage of manufacturers on one hand and users of materials on the other, with a third party of investigators or directors of laboratories between them.

Although these conferences are entirely voluntary, and their resolutions are not binding, but merely the expression of the most advanced and correct opinions bearing on the subjects discussed, it is truly remarkable that they have exerted such lasting influence throughout the civilized world. And this crowning fact is mainly due to Bauschinger's honesty, integrity, and character as a man.

Professor Bauschinger was elected an honorary member of the Society in 1884, and died November 25, 1893.

STEPHEN WILCOX.

Stephen Wilcox was born at Westerly, R. I. His earliest experiments in the mechanical field resulted in the invention of a practical caloric or hot-air engine, previous to Ericsson's first attempt, both men submitting their engines to the Lighthouse Board, by whom they were to be used for operating fog-horn signals.

In 1856, finding that the caloric engine was likely to be limited to small powers, he directed his attention to a form of safety boiler, and invented a set of inclined water tubes connected to an overhead steam and water reservoir, across which the tubes carried the current of hot air from the fire.

It was at this time that, in connection with his life-long friend and partner, Geo. H. Babcock, the Babcock & Wilcox engine was developed, which was manufactured by a number of leading builders, among them the Hope Iron Works, Providence; Morton Poole & Co., Wilmington; Poole & Hunt, Baltimore; and C. & G. Cooper, Mt. Vernon, Ohio. The development of the boiler business, and the fact that at this time the Corliss patents became public property, induced the firm to discontinue the manufacture of this design of engine, although the smaller upright engines of the New York Safety Steam Power Company received a good deal of attention at the hands of Mr. Wilcox.

The Babcock & Wilcox Company became a corporation in 1881, at which time Mr. Wilcox retired from active business life, but kept on making experiments on multiple-expansion engines, with high pressure, in connection with marine boilers, and derived much pleasure from a yacht upon which these ideas were carried out.

In his birthplace, at Westerly, a library building is a monument to his public spirit and generosity; and besides the library and other rooms, it is to contain in its basement a gymnasium for the lads of the town.

He died of pneumonia, November 27, 1893.

JOHN H. HARRIS.

Mr. Harris was born in Troy, N. Y., January 4, 1838. He was a direct descendant of Patrick Henry and of Roger Williams, so

that his ancestry belongs to that which was best in the development of New England character, in combination with that sturdy patriotism which characterized the early patriots of Virginia. His education in Vermont and at Springfield, Mass., was intended to fit him for Yale College, but this plan was given up, and at eighteen years of age he went to sea. He made several voyages to China and Japan, in the last of which he had a narrow escape from shipwreck on the Nova Scotia coast. On his arrival in New York in June, 1861, he at once enlisted as a volunteer in the navy, and while attached to the United States steamships *Albatross*, *Powhatan*, *Vicksburg*, and other minor assignments, he participated in the battle of Port Hudson and in numerous engagements of the squadrons both of the Atlantic and of the Gulf.

Honorably discharged in 1866, he spent a couple of years in the study of law under Senator Hoar, of Worcester; but the routine was disagreeable, and he entered mercantile life. He was in charge of the New York office of the George F. Blake Manufacturing Company; was connected with the Deane Steam Pump Company, and afterwards organized the Harris Steam Pumping Company.

He has, however, become best known to engineering from his connection with the Worthington Pumping Engine Company, as vice-president and general manager of the branch in Great Britain, lending his immense energy and business efficiency to the advancement of the interests of the firm both in Europe and in this country.

He connected himself with the Society in 1890, but for the last two years of his life has been a great sufferer from the progress of the disease which, although alleviated by a most skilful and successful operation, yet was ultimately the cause of his death.

He passed away in this city, January 22, 1894.

SETH BATEMAN WEAVER.

Mr. Weaver was born at Newport, R. I., September 23, 1858. He graduated from Worcester Institute in June, 1881, and worked afterward as journeyman in shops operated in connection with the institute. In 1882 he became journeyman toolmaker with the Crompton Iron Works, and later in same year became inspector of finished piece-work with the Deane Steam

Pump Works, of Holyoke, Mass. He remained with this company as chief draughtsman until January, 1884, when he took the position of chief draughtsman of the elevator department of the Crane Brothers Company, of Chicago, which position he held at the time of his death, February 22, 1894. He was to have been made superintendent of the entire elevator works, had he recovered from the severe operation which was the immediate cause of his death. Aneurism of an artery had followed an accidental discharge of a gun nineteen years ago; but, in spite of this limitation, he had done the work of more than one man, not only in the drawing-room, but as patent expert for his company. Hemorrhage set in nine days after the successful amputation of his limb.

He joined the Society at its June meeting, Providence, 1891.

EDWARD BARRY WALL.

Mr. Wall was born in Kingston, N. Y., April 25, 1856. He graduated with the degree of M.E. from Stevens Institute of Technology, at Hoboken, N. J., in 1876, in the first large class turned out by that institute, and the fourth graduated from it.

The ability displayed in his graduating thesis, on the subject of car framing, led to his appointment under instruction, from July, 1876, to June, 1879, to the Altoona shops of the Pennsylvania Railroad. From June 29, 1879, to January 1, 1882, he was in charge of special investigations and tests, and an inspector of materials for that company, and assistant to the general foreman of the Altoona car-shops, with experience as a designer of cars.

He acted as assistant master mechanic of the Columbus shops of the P., C. & St. L. Ry. until November 1, 1882, and was assistant superintendent of motive power until June 1, 1883, when he was appointed superintendent of motive power of the southwest system of the Pennsylvania lines, with headquarters at Columbus. In April, 1893, he was detailed to act as assistant to the first vice-president, with headquarters at Chicago, and with special charge of the interests of the Pennsylvania lines in connection with the World's Columbian Exposition, after the close of which he was appointed assistant to the general manager of the lines west of Pittsburg, with headquarters at Pittsburg.

Mr. Wall was a notable illustration of the responsibility borne

by young men of this country. His appointment, at the age of twenty-seven, to a superintendency, has been referred to as probably the youngest age at which any man was ever appointed to so responsible a position in an old-established company. His best-known public service was in connection with the work of the Master Car Builders' Association upon the automatic coupler question. It has been said that the advanced status of practice to-day in these matters is due to him more than to any other one man.

He connected himself with the Society at the Pittsburg meeting in June, 1884. His death, April 1, 1894, was due to peritonitis and shock, following an operation for appendicitis. He was universally liked and respected, and his untimely death is deeply regretted.

GEORGE SELDEN.

Mr. Selden was born September 27, 1827, in the city of Erie, Pa. In 1844 he went to sea as a sailor in the East India trade, which at that time compelled a three years' absence from his home. On his return he was among the first to go from Erie to the California gold fields in the winter of 1848-49. On his return to Erie after this experience he entered business with Col. John H. Bliss, in the manufacture of barrels, but the business received a great advance from the development of oil in Pennsylvania, out of which grew the Erie City Iron Works, an establishment employing over seven hundred men, and perhaps the largest establishment devoted to boilers as a specialty in the United States, if not in the world. The success of this enterprise permitted Mr. Selden to indulge his taste for travel, and his collection of curiosities from all parts of the world was very interesting.

He was an active member of the church with which he had connected himself in 1858, just after his marriage, and in which he had held office for many years. His benefactions have been called princely.

The records of the Patent Office from 1869 to 1892 contain many patents issued to Mr. Selden; at first for saw-mill machinery, later for special machinery for oil work, and latterly for details of boilers and engines.

He connected himself with the Society in the Associate grade at the Scranton meeting in the autumn of 1888, and died May 28, 1894.

O. A. LANPHEAR.

Mr. Lanphear entered the profession of engineering from the practical side; he became an apprentice in Cleveland in 1868, passing through the pattern and blacksmith shop, and finally intrusted with erection, both inside and outside of the shop. In 1873 he took charge of the works of the Humboldt Manufacturing Company, in Kansas, builders of iron bridges, and in 1876 he moved to Logansport, Indiana, where he was engaged in building turbine water-wheels and erecting machinery. From 1878 to 1884 he was obtaining his theoretical education, first at Franklyn College, Indiana, and later at the State University at Columbus, Ohio. On graduation he became a member of the firm of Minot and Lanphear, and in 1885, with his partner, entered the Columbus Buggy Company, and for several years served as foreman of the machine shop. The latter years of his life were spent in the general management of a Cincinnati company making a specialty of sanitary appliances.

He connected himself with the Society at the New York meeting of 1886, and died May 24, 1894, from an accident due to a runaway team.

WALLACE H. DODGE.

Mr. Dodge was born July 10, 1849, at Mishawaka, Indiana. He entered his father's store as bookkeeper after completing his education at the University of Notre Dame in his native State. In connection with his business he learned the trade of machinist and copper-smelter and tinner. In 1878 he commenced manufacturing in the lines of wood specialties and hardware. In 1881, in connection with George Philion, his superintendent, the first experimental wooden split pulleys were manufactured, which since 1884 have been so widely distributed under the trade name of the Independence Wood Split Pulley. In 1886 the Dodge system of transmission of power by round ropes was devised, the characteristic being a continuous rope making several turns around driver and driven pulley, with a take-up device at the end of the series of wheels, which this continuous rope enveloped. The young enterprise of the Dodge Manufacturing Company was burned to the ground in 1881 by a lightning stroke, but his energy, ability, and reputation succeeded in building up the business which survives him. Since his return from Europe with the party

of American engineers in 1889, Mr. Dodge has been in ill health, which has compelled absence from home for treatment, but his influence in his native town was not unfelt, and it is to his energy that the introduction of water works and an electric lighting system is credited.

He became a member of the Society in the Associate grade at the Erie meeting of 1889, and his death, on September 10, 1894, was due to neuralgia of the abdomen.

EMIL J. FLACH.

Mr. Flach was born April 28, 1864, in Sweden. He graduated from the Swedish State College at Upsala, whence he entered the Royal Polytechnic Institute in Stockholm, from which he graduated in 1888. He served an apprenticeship at the works of the Stafsjo Engineering and Gun Manufacturing Company, and later attached himself to the Bergsund Engine and Shipbuilding Company at Stockholm.

He had service at sea on the two Swedish steamers *Motala* and *Bergsund*, and after that served in the Maxim-Nordenfeld Gun and Ammunition Company.

Coming to the United States, he attached himself first to the Columbian Iron Works and Dry Dock Company, at Baltimore, Md., and later to the Pennsylvania Steel Company, at their Sparrows Point Works. In July, 1890, Mr. Flach entered the service of the United States Government as draughtsman in the Bureau of Steam Engineering of the Navy Department, and it was upon the recommendation of Commodore Melville that Mr. Flach was retained for the responsible position of chief draughtsman with the S. L. Moore and Sons Company of Elizabeth, and this position is the one held by Mr. Flach at the time he joined this Society, at the New York meeting of 1891. He retained it until he was appointed representative from Sweden to be present at the experiments made with the John Ericsson submarine gun. His commission resulted in an elaborate report to the Swedish Government, and its author returned to the old country.

In November, 1892, Mr. Flach became assistant engineer of the Swedish "Neptune Salvage Company," and took part in the work of raising H. M. S. *Howe*, sunk at Ferrol. It was in an attempt connected with this submarine undertaking that Mr. Flach overtaxed his strength, and contracted the malady which ended his life but a few weeks later. He was obliged to place

himself upon the sick list only a few days before the successful raising of the *Howe*, and although all possible care was taken, no signs of improvement appeared. He was sent in June to the Marine Hospital in Plymouth, England, and from thence to his home in Sweden, installing himself at the Sanatorium at Morsil.

His death occurred at this place, September 2, 1893.

WILLIAM HENRY PATTON.

Mr. Patton was born July 7, 1831, at Princeton, N. J. After receiving the degree of Civil and Mechanical Engineer, at the University of New York, Mr. Patton devoted himself to mining and quartz milling from 1852 to 1886. From 1866 to 1874 he was constructing engineer for a number of silver and gold mining companies in Nevada and California, among them the Old Colony, Keystone, the Plumes-Eureka, etc. He was also assistant superintendent of construction of the United States Mint at San Francisco, and from 1874 to 1877 was constructing engineer for the syndicate Mackay, Fair, Flood, and O'Brien, in Virginia City. From 1877 to 1882 he was their mining engineer, and at the time of his death was mining engineer with the Broken Hill mines, in New South Wales.

Mr. Patton's first introduction to public notice on the Comstock was as chief engineer in the service of the Pacific Mill and Mining Company. While in that position he had much to do in the construction of two mills for this company, the Consolidated and California mills, the largest mills ever erected in the State of Nevada. These mills were run by powerful compound condensing engines, which were devised in part by him, and put in place under his immediate supervision. He supervised the draughting and putting in place of the powerful machinery for pumping at the Consolidated and California shaft, and at the Sierra Nevada, Mexican, and Union shafts; and he was frequently consulted by mining superintendents in regard to the construction of machinery at their mines. He claimed the credit of introducing on the Comstock the method (then new) of deep pumping by direct action with the Cornish system of pumps. He also planned the running of the California mill by wire-rope transmission of water power under a 1,500-foot pressure, and superintended the carrying out of the plan.

He was elected superintendent of the Consolidated Virginia and California mines in the year 1878, when the prospects of these two mines were somewhat on the wane, and he continued to fill that office until he left in the latter part of 1887 to take charge of the Broken Hill mine in Australia. He developed the same capacity as a mining engineer as he had shown as a mechanical engineer. The most notable event in his career as mining superintendent was the finding and developing, after diligent and systematic working, of an important body of ore in the Consolidated Virginia and California mines, which placed those mines on a dividend-paying basis and restored public confidence in the Comstock lode.

In the month of December, 1886, a fire, which had been smouldering since the year 1881, broke out in the timbers of the old stopes of the California mine on the 1,600-foot level, and threatened a cessation of the working of the mine. By a masterly use of mechanical appliances in injecting carbonic acid gas through the burning section for a period of four months, he succeeded in extinguishing the fire; and this enabled the work of extracting ore to be resumed, which continued for some time with profitable results.

Mr. Patton was an ardent lover and devoted student of physics and mechanical science. Although of a naturally modest and retiring disposition, he always came to the front in case of an emergency. Well remembered is his heroic action when, at the risk of his own life, he took the lead of a band of brave men and rescued some miners who were being walled in, in a drift on the lower level of the Alta mine, by a strong flow of water which was beginning to fill the mine shaft.

Another event which occurred a few years ago also showed the quality of the man. A fire broke out in the incline shaft of the Gould and Curry mine, and this was accompanied with a caving of the surrounding ground, which imprisoned a number of miners with no apparent chance of exit. Mr. Patton was early on the ground, giving valuable advice as to what should be done to rescue the men, fearlessly taking an active part himself, and directing movements in the face of that terrible catastrophe.

He became a member of the Society at the New York meeting in 1882, and died August 7, 1892, at San Rafael, Cal., of disease of the brain.

JOHN IRWIN VEEDER.

Mr. Veeder was born at Sewickley, Pa., on June 11, 1858, and passed away in the early prime of manhood, on November 20, 1892, at Chicago, Ill.

In 1866 his parents removed to Burksville, Ky. Here his musical education was begun, and at the early age of nine years he performed on the piano at a public concert.

In 1868 the family moved to Sewickley, Pa., and thence, in 1869, to Plattsburg, N. Y. Here, his father conducted the affairs of the Arnold iron mine, in the interest of Dr. C. G. Hussey, Hon. Thomas M. Howe, and himself, for twelve years, the mine having then been sold.

It was during this formative period of the boy's life that he acquired his principal education. He attended the Plattsburg Academy, where he was an eager student, so much so that it was often necessary to call a halt in his studies for a few days, in order to give rest to a constitution never very strong. To this end, he was sent, early in 1872, to Minneapolis, Minn., where he remained a year.

He spent the last four or five months of 1874, at Arnold Hill, down in the mine, in getting a knowledge of the running of drifts and cross-cuts, timbering, pumping, blasting, stoping, use of hand and machine drills, and a great variety of other underground matter. Afterwards, on surface, he spent several months during the winter in running an engine and boiler at a small mine owned by the Company. He used to get up at four o'clock, on the coldest mornings, and walk half a mile, often in the deep snow, to the mine, in order to get up steam in time for beginning work at seven o'clock.

After graduating, in 1876, from the Academy, with highest honors, he took a complete course in Eastman's Business College at Poughkeepsie, N. Y. About this time the selection of a cadet for West Point was in the hands of the Hon. Andrew Williams, M. C., who desired him to compete in the customary examinations for the honor, but his proposed competitors declined to appear if he presented himself. His father, however, wishing to fit him for an earnest, practical business life, declined in behalf of his boy, and gave him some further business experience.

Early in 1877 he went to Chicago, and spent some time in a stationer's store, but had much trouble with his eyes, being blind

for several weeks. He therefore returned to Plattsburg, and spent the last five months of the year at the office and shipping station of the Arnold mine, at Ferrona. Here he spent his mornings in scientific studies, and his afternoons in assisting in the loading of ore upon the cars.

In February, 1878, his father having taken an interest in the Saranac Horse-Nail Company, whose works were also located at Vergennes, Vt., the boy went to Vergennes, and spent several months there in learning the making of horse-nails and the building of the machines. The works were removed in November, 1878, to Plattsburg, N. Y., where all of the proprietors lived; and after the installation and starting of the new plant by Mr. William Long, and the resignation of the latter, Mr. Veeder became foreman of the factory, before he was twenty-one years old. Here, his fine inventive talent, his good judgment of men, his admirable tact, and perfect self-control, made his administration an excellent success.

Owing to changes in the business, he resigned his position, and began to devote himself to a systematic training as a machinist, it then being the thought of his father and himself that he would ultimately devote himself to the machine business. In order to get the best and largest experience, he took positions in various shops, going first to Burlington in February, 1880, where, in the shops of B. S. Nichols & Co., he constructed some new nail machines, under the direction of Mr. Long, for the factory at Plattsburg.

In October, 1881, he went to Montpelier, Vt., where he worked in the shops of the Montpelier Iron Company for a short time, and then to Hartford, Conn., in the shops of the Hartford Engine Company.

Failing health compelled him to give up severe labor, and in December, 1881, he went first to Minneapolis, and then in the following month to the stock farm of Gen. A. Y. Mackay, at Bazile Mills, in Northwestern Nebraska.

In July, 1885, with health much improved by the out-of-door work, he entered, as office boy, the office of James H. Raymond, member of this Society, and a prominent patent lawyer of Chicago. With this modest beginning, he soon proved himself worthy of better things. Mr. Raymond himself paid him the highest possible tribute, in a communication to the Patent Law Association of the city of Chicago.

Music was the chief solace and recreation of his life, and his skill as a violinist was of a very high order. To his musical studies and practice he gave nearly all of his time when out of his office, and he was one of the founders of an amateur symphony society of over sixty members; but the cares of an ever-increasing business, necessarily very exacting in its nature, proved too much for his constitution, which was, from his birth, far from being strong. He connected himself with the Society at the San Francisco meeting in May, 1892.

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